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The Marine Engineer

AND NAVAL ARCHITECT.

Editorial and Publishing Office—
33, Tothill Street, London, S.W. 1.

MARCH, 1923.

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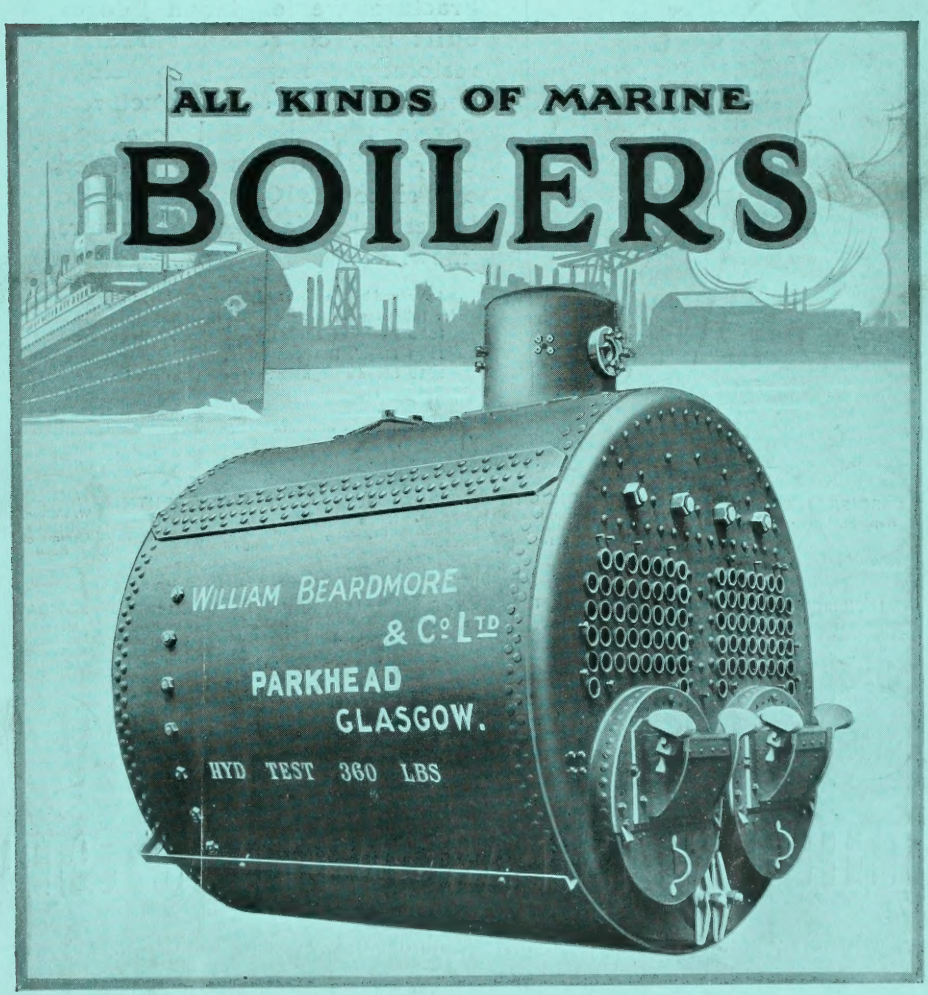

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The Editor will be glad to consider original articles and paragraphs on all subjects indicated by the title. Short practical articles based on the sender's own experience and illustrated by photographs or drawings are desired.

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Sir Alfred Yarrow, Bart., has made a gift of £100,000 to the Royal Society, the money to be devoted to the promotion of scientific investigation and its application to practical affairs.

Major R. A. Jones notifies that, after several years' military service, he has resumed his practice as consulting engineer and surveyor, in South Wales, with offices at Chappell's Chambers, Castle Street, Swansea.

Mr. J. W. H. Young, a Canadian superintendent engineer, has arrived in this country to superintend the building of Canadian Lake steamers at the Wallsend Shipyard of Swan, Hunter & Wigham Richardson Limited.

Ir. J. C. Arkenbout Schokker, principal Assistant for shipbuilding in the Technical University, Delft, has been appointed private teacher for the special subject of "The Propulsion and Movement of Ships at Sea."

Mr. E. A. Gawden has been appointed to the Admiralty inspection staff at Birkenhead in connection with the building of the new capital ship there. He has been on the Admiralty Overseer's staff at Barrow in-Furness since 1908.

Mr. H. D. Couche has joined the staff of Lionel S. Taylor & Co., consulting engineers, Royal Liver Building, Liverpool. Mr. Couche is well known on Merseyside, having been for many years chief ship's draughtsman with Cammell, Laird, at Birkenhead.

Mr. E. J. Vivian has been appointed Assistant to the Engineer Overseer at Birkenhead in connection with the construction of the new battleship *Rodney*. Mr. Vivyan has been stationed at Birkenhead on behalf of the Admiralty since the building of the battleship *Audacious*, which was completed in 1912.

Mr. J. Y. M. Kennedy, son of Sir Alexander M. Kennedy, Chairman and Managing Director of Northumberland Shipbuilding Company, who, for some years, has acted as Assistant to Mr. W. J. Paulin, Managing Director of Donkin & Co. Limited, Walkergate, Newcastle, has been appointed a Director of the firm.

Mr. G. F. Dowding has resigned the London Managership of the Talbot-Stead Tube Company Limited and is continuing his engineering business at *Morning Post* Building, Strand, W.C., and also as London Sales Manager of the Projectile and Engineering Company Limited, and the Sheffield Twist Drill and Steel Company.

The Council of the Institution of Electrical Engineers have made the second award of the Faraday Medal to the Honourable Sir Charles Algernon Parsons, K.C.B., F.R.S., Honorary Member of the Institution. The medal is awarded for "notable scientific or industrial achievement in electrical engineering, or for conspicuous service rendered to the advancement of electrical science."

The death occurred on January 8 of Professor S. Terano, D.Eng., M.I.N.A. He was born in November, 1868, and was educated as a Naval Architect in the Imperial University, Tokyo, where he graduated in 1890. He also studied for some time at Glasgow University. He was Principal of the Aeronautical Research Laboratory of the Imperial University, President of the Japan Society of Naval Architects, Chairman of Directors of the Imperial Japanese Corporation, and an active and prominent member of several Government and Public Committees.

The death has occurred in London of Mr. Adam Scott, M.Inst.C.E., at the age of 68 years. He was well known in connection with the construction of large harbour and dock engineering works and was the Admiralty superintending civil engineer on the harbour works and dockyard at Gibraltar, which were begun in 1893 and finished about 1906. He had also been connected with important work on the Weaver Navigation, the Clyde, Tyne, and Thames, and also in Brazil and Chile. Mr. Scott was the engineer for the work in connection with the enlargement of the Deptford Graving Dock at the shipyard of Sir James Laing & Sons Limited, Sunderland.

The Council of the Institution of Naval Architects have awarded the Gold Medal for 1922 to Mr. J. L. Kent of the William Froude Experimental Tank for his paper "Experiments on Mercantile Ship Models in Waves," and Premiums have been awarded to Mr. E. Leslie Champness, M.B.E., M.Sc., for his paper "Longitudinal Strength of Cargo Vessels and its Variation with Fullness of Form," and Major L. Fea, of the Royal Italian Navy, for his paper "Some of the Consequences of the Washington Conference with Regard to Naval Construction." The above awards will be presented on March 21, at the opening of the annual meetings, which will be held at the Royal United Service Institution, Whitehall.

EDITORIAL.

THE COMBINATION OF THE TALENTS.

UNDER the heading "The Battle of the Talents" we considered, in an editorial in our January number, the industrial conflict which at times rages between the opposing forces of the theory-led and the practice-driven. It was our opinion, after viewing the field of battle, that the time for the calling of a truce had arrived in order that the Generalissimo of Practice and the Research Board of Theory might get together and consider why they should be opposed to each other when both were very definitely opposed to the common enemy, Industrial Want. We suggested as a solution, that of mutual respect and compromise, each force being made to realise that without the support of the other, it would be swamped by their common enemy. Our solution meant the continuance of two camps and thus duality of control. "Dunelm," in a letter to our February number advocated the unity of the camps, and thus of control. Thus, instead of a battle of talents, a new force comprising the combination of the talents would be available for the onslaught on Industrial Want. "Dunelm's" views are obviously our own a stage further developed; and with further progress in the development of technical education, these views eventually will be realised. Even at the present time, there are quite a number of our leading shipbuilders and engineers, whose technical attainments have undoubtedly assisted and moulded their practice, and are ample evidence of the truth of "Dunelm's" view. On the other hand, however, we regret that our own view is also, only too true. In spite of technique, the old school of rule of thumb still holds many key positions, and our view was that these practical, work-loving men, should be persuaded to listen to reasoning, based upon a scientific analysis of their own experience and that derived from countless other and entirely unexpected sources. This analysis is, of course, what we have learned to regard as theory. The so-called practical man in his haste to do and to make, relies, unconsciously, upon his subconscious mind to carry out this analysis; the lessons of his experience are entrusted to his muscular memory. Thus engineering and shipbuilding is, in all too many cases, not a question of brains, but merely one of muscle.

It is in this analysis of experience on the part of the practical man that the *battle* of the talents must certainly give place to the *combination* of the talents. At present, as we pointed out, the assistance of these men who have been especially trained to think and to analyse must be made use of by the practical man, for it is very questionable whether the practical man could or would really himself analyse his own experience even if the opportunity presented itself. For instance, in periods of industrial depression, we have heard it suggested that a whole drawing office staff should be turned on to the analysis of a firm's experience, to consolidate its data and to examine its practice at the only time when perspective is possible. This may have been done during the past year by some firms, but in one shipbuilding firm, not small by any means, draughtsmen were retained merely to clean up old models and to ink in old "lines" which, in the "fat" years, had been left in pencil! This has been this particular firm's practice for more years than we care to mention. We merely give this example to show that a stronger case exists here for the combination of the talents in the form of enlightened management than could be the case with any form of compromise, owing to this particular firm's previous wealth of experience. That technical education will eventually dispose of the battle of the talents will be generally admitted. But technical education in itself requires considerable development. A man's technical education only begins when his school days are over, and three years spent at a university enable him to obtain a sound grasp of the first principles of his profession. He secures intensive learning without much visible sign of mental fatigue. When, however, the young graduate enters industry, his rate of assimilation, almost invariably, and perhaps naturally, diminishes. Very few firms take a definite interest in the practical education of the mass of their apprentices, and thus an obstacle is placed in the way of men who are capable of combining the talents. We would not say that this difficulty is intentionally placed there by the "old school," but nevertheless we feel that not a little persuasion would be necessary in order to secure its removal. There are, however, signs that some firms are paying more attention to the works' education of apprentices. Dr. Lippani, in a recent lecture before the V. d. I.* dealt very ably with this aspect of the subject, and a perusal of the report of his lecture shows that, in Germany at any rate, the battle is giving place to the combination of the talents. In this country, Capt. Riall Sankey, in his recent Presidential Address before the Junior Institute of Engineers, dealt admirably with the "Utility of Theory to the Practical Man," and put forward an excellent appeal for the combination of the talents. Dr. Hatfield, speaking at the discussion of Dr. Mellanby's recent North-East Coast paper on "Turbine Failures," directly accused engineers of being lacking in technique. The remedy he proposed was not, as might be supposed, the wholesale employment of metallurgists by engineers, but that engineers might themselves endeavour to learn more about the metallurgy of the materials they use. Dr. Hatfield thus recommends the combination of the talents. Thus even now there are signs that the battle of the talents is on the wane, and that, in the future, industry will be developed by those who either in themselves are a combination of talents, or who, like some of our most successful business men, know how harmoniously to combine the talents of others—an asset which is unquestionably one of the most fundamental principles of organisation.

GAS TURBINE DEVELOPMENT.

THE fact that a 5,000 kw. Holzwarth type gas turbine is now approaching completion in the works of Messrs. Thyssen, at Mulheim, is likely to form a highly important stage in the development of the internal-combustion engine, for should this machine develop its designed power during the trials for which it is so nearly ready, it will undoubtedly be the most powerful individual internal-combustion engine yet built, the record output thus passing from the well-tried and certainly much more efficient reciprocator, to a type with which comparatively little experience has yet been gained. Herr Holzwarth, as is well known, has been working on his design of gas turbine since 1908, and has previously constructed a vertical machine of about 900 h.p. with which very extensive experiments have been conducted, as well as, more recently, a 500 h.p. horizontal oil-combustion turbine. As the result of experience with these machines, the 5,000 kw. unit has been constructed, while designs for very much larger machines have been prepared in great detail.

The type of turbine, however, which Herr Holzwarth prefers, is designed to operate on the constant-volume cycle instead of on the constant-pressure cycle on which nearly all other experimental internal-combustion turbines have been made. These have all been of very small size, Armengand's second machine, which is the largest hitherto built to work on the constant-pressure cycle being only about 300 h.p. Some very valuable information has been elicited as a result of the many trials of the Holzwarth machine which must certainly prove to be of great assistance to other designers in their efforts to solve the gas turbine problem, which is, perhaps to an exaggerated extent, supposed by many to depend mainly on the provision of new materials capable of withstanding high temperature. What Holzwarth has done at the least is to prove that

* Vereines deutcher Ingenieure, December 16, 1922.

gas turbine blading can be made to stand an impinging temperature of nearly 900° Fahr., and, though this is no more than has been done by other experimenters, none have hitherto developed practical commercial machines working under such circumstances. But the very success attained in the Holzwarth design opens up a question that seems destined to create in the future some such degree of discussion as the relative merits of the four-cycle or two-cycle oil engine have already provided, and this will, as far as the internal-combustion turbine is concerned, be largely directed towards the nature of the cycle to be adopted. Both types present advantages and disadvantages compared with the other, but a curious feature of the constant-volume type lies in the fact that the complicated valve mechanism which is accepted almost without question in the case of the oil engine and the petrol motor is precisely the feature which is condemned in the case of the turbine. It is true that it is always possible for engineers to accept a disadvantage in exchange for a great gain, which latter, in the oil engine, is undisputedly its high thermal efficiency, but a valve mechanism which will operate in the one should certainly do so equally well in the case of the other, particularly when neither pressures nor temperatures attain such high values as they do in the Diesel type of engine.

In practice neither the constant-volume nor the constant-pressure cycle is found in its pure form, for a single engine may embrace characteristics falling into both categories. Even the Diesel engine does not work altogether entirely on the constant-pressure cycle; the earliest portion of the change is burnt under conditions of constant volume. The more nearly maximum pressure and maximum temperature coincide, the more nearly does the cycle approach the constant-volume condition. As the heat rejection usually takes place at a pressure slightly above atmosphere, the constant-pressure cycle will not work unless the air supply in the combustion chamber is under pressure, and the efficiency of the cycle depends on this pressure increasing with the compression ratio. It is, of course, possible as an alternative to have actual compression above atmosphere, to reject the heat at a pressure below atmosphere by using an exhauster to withdraw the products of combustion and deliver them to the atmosphere, and this method possesses considerable advantages, although it does not yet appear to have been tried. The constant-volume cycle can, if necessary, be used without any pre-compression at all, and, in fact, the earliest gas engines worked without any. Nevertheless, compression greatly increases the possible efficiency of a cycle by increasing the range of expansion, just as in the steam turbine the range was extended by the use of vacua beyond the scope of the piston engine.

In the case of the gas turbine, where either compression or exhaustion forms the negative work of the cycle, the choice lies between the simplicity of the constant-pressure type of machine, with its relatively large compressor, or the constant-volume type with very small compressor and complicated valve mechanism—mechanism which is, after all, practically identical with that so widely used in the oil engine. With the constant-volume cycle, the combustion chamber works intermittently as in the piston engine, and Holzwarth has used six chambers in his oil-combustion turbine, and eight in his big gas turbine. The objection is that the velocity of the gases leaving the nozzles after the explosion has occurred is a variable quantity, diminishing as it does with the fall of temperature in the explosion chamber, although Holzwarth denies that this has a detrimental effect on the efficiency, a claim which it is somewhat difficult to believe in view of the well-known laws governing the efficiency of turbine blading. One advantage to offset this defect lies in the case of scavenging and the cooling effect of the scavenge air on the turbine blading. In the constant-pressure turbine, there are no valves whatever. The fuel is forced into the furnace under pressure and the only moving element in the machine subject to temperature, and that only to the exhaust gases, is the rotor. Against this unique simplicity, lies the fact that the big compressor may take up so much of the work of the turbine that insufficient is left over to render the remaining positive work efficient. So little work of compression or exhaustion is needed in the constant-volume cycle, that the power required to drive them can easily be obtained by steam regenerated by the exhaust gases from the turbine; the quantity of the heat thus available is quite insufficient for the larger compressor necessary for the constant-pressure cycle.

With low compression in the constant-volume cycle it becomes less easy to burn satisfactorily heavy oil as fuel, although Holzwarth has undoubtedly done so. With power gas, this difficulty vanishes to a great extent. But in the constant-volume turbine, ignition gear is required, whereas, once lighted, the fuel combustion in the constant-pressure turbine furnace is a continuous process. For marine work it must be admitted that there is no apparent future for the constant-volume type of turbine. It has been severly criticised on account of the mechanical complication of its valve gear, although that is somewhat unjust in view of the wide use of similar valves in motorships. It would also seem that one fundamental ground of preference for one type of cycle or the other would lie on the side of the machine with the highest efficiency. Now, the impinging temperature on the turbine wheel forms an insuperable limit at this point of the cycle, nor can the blade speed in a given temperature be any greater in one case than the other. Starting from this point, for the some quantity of work done on the turbine wheel for the same quantity of heat supplied, *i.e.*, at the same efficiency, the constant-pressure cycle machine will require a much higher compression ratio. The constant-volume cycle has a much greater ratio of positive to negative work, an advantage which must not be underestimated. As regards radiation losses, that in the constant-volume explosion chambers can be brought down to a very low point—lower probably than in the piston engine, and much lower than in the constant-pressure turbine. The consequence is that there will not be a very great deal of difference in the brake thermal efficiency of the two types of turbine. Hitherto the Holzwarth tests are rather obscure. The claim is made for a total efficiency of 25 per cent., a figure which is far superior to that reached with the early experiments in the constant-pressure turbines. It, is, however, on the high side of what can be expected from calculation, though a properly designed constant-pressure machine might obtain this figure provided that the compressor efficiency did not fall below 70 per cent. compared with the isothermal. It is obvious that such a figure as even 25 per cent. is far below the 32-36 per cent. obtained with heavy oil or Still engines, but it must be remembered that hitherto work on the gas turbine has been very restricted, and until temperature limits can be raised by the employment of new materials, efficiencies much exceeding 25 per cent. are out of the question. On the other hand, the turbine unit can be built in almost any power per shaft, and the weight of an installation, even with generators and shaft motors, for it is in the electric-drive that its only chance of adoption aboard ship will lie, will be very considerably less than that of the large reciprocating unit.

High-speed Pleasure Launch.—John I. Thornycroft & Co. Limited, recently completed a high-speed touring launch for His Highness Prince George Lotfallah, Cairo. The boat has a length of 40 ft. with 8 ft. beam and a draught of 2 ft., and is wall-sided with V-shaped bottom sections. The hull is constructed of teak and is sheathed to well above the waterline. The engines, which are installed immediately forward of the steering position, consist of two Thornycroft six-cylinder Z.6 type marine petrol-driven motors which develop 70 b.h.p. at about 1,374 r.p.m. The launch recently ran successful trials on the Thames when she comfortably attained a speed of over 21 knots.

BOOKS FOR THE MARINE ENGINEER AND NAVAL ARCHITECT.

Brief Reviews of Recent Publications.

A New Dutch Publication.—The first number of a new half-monthly publication entitled *Sterkstroom* has been forwarded to us. It is published by the proprietors of *Het Schip*, which latter is well known in shipbuilding circles. The periodical is intended to deal mainly with matters connected with electricity.

Internal-Combustion Engines.—A rechauffé of papers read and discussed at meetings of the Institute of Marine Engineers during the last three years. Published by the Institute of Marine Engineers, at 85, The Minories, E.1. 6 in. × 8½ in. × 1¼ in. 595 pp. Price 12s. 6d. net. Illustrations and diagrams.

This book contains a collection of 18 papers dealing, with various aspects of the internal-combustion engine, which have been read before the Institute of Marine Engineers. It is a production which should prove of value at the present time when the internal-combustion engine is becoming such an important factor in marine engineering.

Twentieth Century Guide for Marine Engineers. By J. A. Ramsey and J. Rosbloom. London: Chapman & Hall, 11, Henrietta Street, Covent Garden W.C.2. 537 pp. Price 15s. net. Illustrations and diagrams.

This book is an American one, written by two practical marine engineers, one of whom is a warrant machinist in the U.S. Navy and the other of whom is an instructor at the Steam and Internal Combustion Engineering Schools, at the Mare Island Navy Yard. It is a very comprehensive little work for its size, and gives a great deal of useful information of a practical nature without going deeply into the theoretical side of the question. There is a section on the Development of the Steam Engine, followed by others dealing with boilers, piping, valves, pumps and other auxiliaries. Practical engineers will possibly find much of interest in the 100 Questions and Answers on Reciprocating Engines, whilst those interested in turbines, gas engines and Diesel engines will find sections dealing with these types of prime movers. As we have mentioned, the book contains a mass of useful subject matter of a practical nature, and the treatment is simple and straightforward. We feel, nevertheless, that the first section dealing with arithmetic is of a somewhat too elementary nature, and that, if eliminated, the space so gained might have been utilised with advantage by expanding some of the other sections.

Sea-going and other Concrete Ships.—By N. K. Fougner, C.E., M.Am.S.C.E., A.M.Inst.N.A. London: Henry Frowde and Hodder & Stoughton, 1, Bedford Street, Strand, W.C.2. 6¼ in. × 10 in. × 1 in.; 216 + viii pages. Many illustrations. Price 21s. net.

The author has endeavoured in this book to present a true record of sea-going concrete ship progress up to the present time, and in fact, he has mainly drawn from his own personal experiences in the construction of some 30 concrete ships during the last five years. In this country, during 1918, the Ministry of Shipping laid down a programme for concrete ship construction, which contemplated a total of about 150,000 tons d.w., but the Armistice practically saw the complete abandonment of this programme. It is estimated that the total world tonnage of concrete sea-going ships to-day does not exceed 150,000 tons, and the author states that, inasmuch as this tonnage consists of experimental ships built by men with little previous shipbuilding experience, and who have never had an opportunity to build a second vessel, no definite information as to first cost, cargo capacity, speed, running expenses, &c., is so far available. Recommendations are set forth for constructional details, among which are, that nothing is to be gained by departing from the exterior shape as adopted in steel ships of recognised good practice; frames need not be placed so closely as in steel ships; extra depth is not necessary to secure adequate longitudinal strength; launching considerations are in all respects similar to those for steel ships, &c.

Naval Architecture—A First Course. By W. M. Gray, B.Sc. London: Methuen & Co. Limited, 37, Essex Street, W.C. 8¾ in. × 5½ in. × 1¼ in. 201 pp. Price 12s. 6d. net.

A technical work without a preface is rather difficult to review. Mr. Gray's book unfortunately suffers from this defect, and although its title implies that it is especially intended as a first course in the subject, this title, to our mind, hardly describes the contents of the book, which seems to be written especially for evening class and technical college students where the guidance of a lecturer is easily obtained, and the treatment is such that a certain acquaintance with ships and their properties is presumed. The nature of the book is more truly that of a second course than a first, and from this point of view the treatment is very thorough. In reviewing an elementary text-book, the pedagogical point of view is even more important than the technical, and several points of a pedagogical nature could be raised in criticism of Mr. Gray's work. For instance, the sequence of the chapters is hardly logical, and a first course in Naval Architecture which starts with the words "The fore perpendicular is . . ." seems rather lacking in the approach to the subject.

The treatment of approximate integration and displacement calculation is exceptionally thorough and we unhesitatingly congratulate the author on this portion of his work. If anything, however, this portion is probably too detailed, and we would have liked to have seen the author make a case for the adoption of the simplest methods rather than explaining all. There is one chapter in Mr. Gray's book which we consider to be rather out of place in a first or even second course on the subject. We refer to the chapter on "Riveted Joints." It is well known that the elementary theory of riveted joints usually given in text books is quite erroneous and does not even approximately represent the true behaviour of a riveted joint. It is doubtful therefore whether this elementary theory should be perpetuated, for it will most certainly have to be unlearned by the student before he proceeds to the higher branches of the subject.

Several other minor points of a pedagogical nature could be mentioned, for when so many excellent elementary text-books exist on the subject, any new book must possess considerable educational advantages in order to justify its existence. However, so far as detail and clarity of treatment is concerned, Mr. Gray's book has much to commend it. It is a good exposition of what is usually considered elementary naval architecture and should form a useful text-book for evening class and technical college students, who will find in it a helpful guide to the practical theory which they will ultimately have to apply as draughtsmen and designers. The text is well illustrated both by diagrams and well-chosen numerical examples. With the appearance of a second edition of Mr. Gray's work, we trust that a preface will be included, in which any necessary personal advice for the guidance of the young student can be profitably embraced.

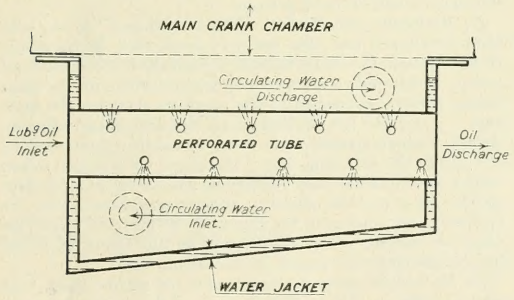
EMERGENCY ENGINEERING ABROAD.

An Account of Quick Repair Work executed under trying circumstances in the Near East.

The repairs to the fleet auxiliaries assisting at the evacuation of Chanak, as a Correspondent pointed out in last month's issue, were undertaken by the British light cruiser *Centaur*, which ship also acted as evacuation vessel. To assist in the evacuation, a fleet of between 20 and 30 nondescript craft had qeen hurried down from Constantinople. These craft varied from fast 70 h.p. twin-screw motor-boats to the most ill-conditioned tugs. As they were manned by a very cosmopolitan crew, the task of keeping them in running order was rendered all the more difficult for the naval engine-room ratings. It must be pointed out that the cruiser *Centaur* possessed no foundry, and that her engine-room stores were necessarily limited to ship's requirements.

A Motor-Boat Repair.

The first repair was of interest. A 70 h.p. motor-boat reported that the starboard engine was running hot, and out of action; and that by some means salt water had become mixed with the oil used in the forced lubrication system. The boat was ordered alongside the cruiser, and the starboard engine disconnected and hoisted out. It was decided to remove and test the oil cooling tank, which was attached under the main crank chamber as shown below.

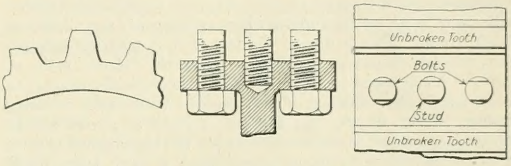


Sketch showing Oil-Cooling Tank in Position.

The cooling-chamber was of copper, provided with a water jacket and cooling water inlet and discharge, as shown. Through this chamber a perforated pipe was fitted, through which the cool oil entered the crank chamber proper. A water-pressure test of 10 lb. per sq. in. was applied to the jacket, and it was found that, owing to the number of leaks, a new lning would be required. As the outer and inner skins were secured by brass rivets the job necessitated cutting the whole of these out and turning up new ones. A new copper inner skin was made, using $\frac{3}{64}$-in. sheet, placed in position and riveted up. As an additional safeguard, all the corners and rivet ends were well soldered over, additional metal being used to make shoulders where the oil pipe passed through the two skins. Before connecting up to crank chamber a water-pressure test of 20 lb. per sq. in. was applied, with satisfactory results. The chamber was then connected up, and within 12 hours from taking in hand the defect the boat had run a satisfactory full-power trial.

Repairing a Toothed Wheel.

A tug carried away several teeth of the winch main-driving wheel, and it was essential that the repair should be quickly carried out, as the winch was called upon to perform exceptionally heavy work. The ordinary studding repair method was considered insufficient. The broken teeth are shown below, and also the section of rim wheel. The broken parts were first chipped and filed flush to the root and a central stud fitted as shown. On either side holes were drilled right through the rim. These holes being tapped. Long bolts were now fitted, projecting as shown, these being as great in



Sketch showing Method of Repairing Broken Gear-Wheel Teeth.

diameter as the body of wheel would stand. These bolts were made a tight fit. A template was next made to fit three unbroken teeth, and the stud and bolts were roughly filed to conform to this. The plan of the rim with repair completed is also shown. Several wheels of the same section were treated in a similar manner, and as far as could be ascertained no trouble was experienced. What was of the utmost importance, this repair took only two hours to complete.

New Propellers.

As it was frequently necessary to order new propellers, it was essential to find the pitch. No elaborate shop appliances being at hand, the following methods were adopted. When it was possible to remove propeller, a wooden disc was turned to fit propeller boss, and a wooden batten fixed cross-wise, so that the batten could swing round with the axis of propeller boss as centre, and in a plane perpendicular to the axis of boss (see diagram). The

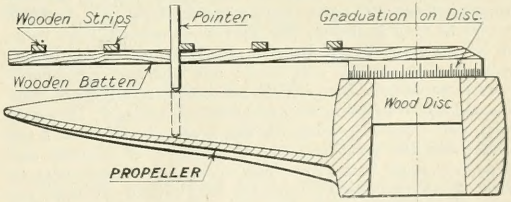


Diagram showing Apparatus Improvised to Measure Pitch of Propeller.

wooden disc was graduated round the circumference, generally every 7½ degrees. Small strips of wood were now fastened on the batten at known distances from the centre, and a pointer held vertically against these strips. The batten was swung round through a division, and the difference in height between batten and face of the blade was measured by the aid of the pointer. If the batten was swung through 30 degrees, the measurements made in inches, the number of inches represents the pitch of propeller in feet. The measurements are made at different radii and the average taken.

When the propeller was in place on the shaft, the batten was fastened to a cross-piece shaped to suit the curvature of any suitable boss or flange instead of the cylindrical portion as previously described, and the same routine gone through.

REDUCTION GEARING FOR STEAM TURBINES.

By CAPT. CHAS. W. REED, B.Sc.

Gearing is a necessity for slow speed, and advisable for high-speed turbine-driven vessels, in order to obtain the greatest efficiency of both the turbines and the propeller, while at the same time keeping the size of the turbine within reasonable limits. For an average 10 to 12 knot merchant vessel, the maximum propeller efficiency is obtained at the comparatively low speed of 70 to 80 r.p.m., while the turbine efficiency is at its maximum when running at, say, 2,000 to 3,000 r.p.m. This is due to the fact that a reaction turbine, say, runs at its maximum efficiency when the velocity ratio, or ratio of blade speed to steam speed, is theoretically nearest to unity, and the efficiency drops rapidly with lower velocity ratios. Then, as the steam speed is fixed at about 400 to 600 ft. per second, it follows that to get the blade speed near to this, with low revolutions, necessitates a very large blade ring diameter of the turbine.

For example, taking a turbine running at 80 r.p.m., to obtain a V.R. (velocity ratio) of even 0·9, needs a mean diameter of about 129 ft., which is, of course, impossible.

$$\text{Blade speed} = 0{\cdot}9 \text{ steam speed} = 0{\cdot}9 \times 600 = 540.$$

$$\text{Diameter in feet} = \frac{\text{Blade speed} \times 60.}{\pi \times \text{Revs. per min.}}$$

$$\frac{540 \times 60}{\pi \times 80} = 129 \text{ ft.}$$

Now, if we increase the revolutions to 3,000, the diameter is reduced to 41½ in. On the other hand, if we keep the diameter within a reasonable limit of, say, 6 ft., and allow revolutions of, say, 400 per minute, we get—

$$\text{Blade speed} = \frac{\pi \times 6 \times 400}{60} = 126 \text{ ft. per second,}$$

which gives a velocity ratio of only 0·21, and a very considerable loss of efficiency in the installation.

Before gearing was introduced, a compromise had to be effected (in high-speed vessels) by having direct-driven turbines running at, say, 400 r.p.m., but, as shown above, these turbines would be very inefficient (the maximum velocity ratio being in the region of ·5) and become very large, costly and heavy, even for fast warships and liners; and with slow-speed cargo vessels would have been of such a size as to render them quite incapable of competing with reciprocating engines. However, with well-designed gearing, the loss need only be about 2 per cent., and the great increase of efficiency of turbines and propeller (about 15 to 20 per cent.) more than counter-balances this. Single-reduction gearing has been extensively employed in the Royal Navy, but with slow-speed cargo vessels, double-reduction gearing allows a larger reduction with smaller wheels, and, consequently, is more in use. Another advantage of having gearing is that high-speed steam turbines can make better use of superheat and high vacua, with a corresponding increase in efficiency, the use of 100° Fahr. of superheat reducing the steam consumption by about 10 per cent. Vacuum increased from 28 in. to 29 in., reduces the consumption by about 6 per cent., whereas an increase of 10 lbs. per square inch in boiler pressure only reduces consumption by about 1 per cent. A comparison between a battleship having direct turbines with a steam consumption of 12 lbs. per shaft horse-power per hour, and one having single-reduction geared turbines and 150° Fahr. superheat, with only 8·7 lbs. per shaft horse-power per hour, shows the saving which may be realised with gearing.

Types of Gearing.

There are three chief types of reduction gearing in use, as follows:—

(1) Electric.
(2) Hydraulic.
(3) Mechanical.

(1) Electric transmission is performed by turbo-generators supplying current to electric motors connected to propeller shafts, and is in favour in the United States Navy, but is not greatly used in this country.

The advantages claimed are:—(1) At low powers only part of the plant need be used, for example, one turbo-generator may be run at full speed in place of two at low power and efficiency; (2) No need for separate reversing turbines, as the motors only need be reversed; (3) Full power astern is available, in place of the usual 60 to 70 per cent.; (4) The generators may be placed where convenient, for example, on a flat.

However, the above advantages are much more important in Naval than in merchant vessels. Moreover, electric drive is usually heavier and more expensive, while being lower in efficiency, about 90 to 92 per cent.

(2) Hydraulic transmission. The "Fottinger" is the only type developed, and this has only been used in Germany. In this system, the steam turbine is connected to a centrifugal pump, which delivers water to a water-turbine in the same casing, directly connected to the propeller shaft on the same axis. A similar but smaller pump delivers water when required, through special guide vanes to another water-turbine and forms the reversing unit; therefore, no separate astern turbine need be fitted, and the turbine may be run at a constant speed. However, the efficiency of the transmitter is low, the maximum only being in the region of 80 per cent., and this drops off rapidly when not running at full speed, or if the ratio of gearing be increased beyond about 6 to 1.

(3) Mechanical gearing. Spur gears for marine work were used as early as 1853 in H.M.S. *Duke of Wellington*, but were increasing gears (in the ratio of 2¼ to 1) to permit of the old type slow-speed engine driving the higher-speeded propeller. The present gearing is, of course, reducing gear, and with double-reduction, gives a ratio as high as 40 to 1.

All the marine gearing now in use is of the double-helical type. Involute contour teeth are used, as with this tooth the uniform velocity transmission remains constant with slight alterations of centre-distance between wheel and pinion shafts, due to clearances and wear of bearings, flexibility of gear-case, unavoidable alignment errors, &c. The efficiency with well-cut teeth is very high, the loss only being about, 1½ per cent. with single, and 3 per cent. with double-reduction gear. Therefore, as mechanical gearing is the most efficient and most extensively used, we will deal more fully with this type.

Gear Mounting and Design.

There are two distinct types of gear cases in use, the heavy rigid type and the floating-frame type. The latter is used largely in United States of America, and consists of mounting the pinions and bearings in a frame supported on a flexible "I" beam, which is claimed automatically to maintain its correct position after being first accurately aligned. Nevertheless, it is not in favour in this country, as it appears that its advantages are outweighed by its disadvantages. It is essential

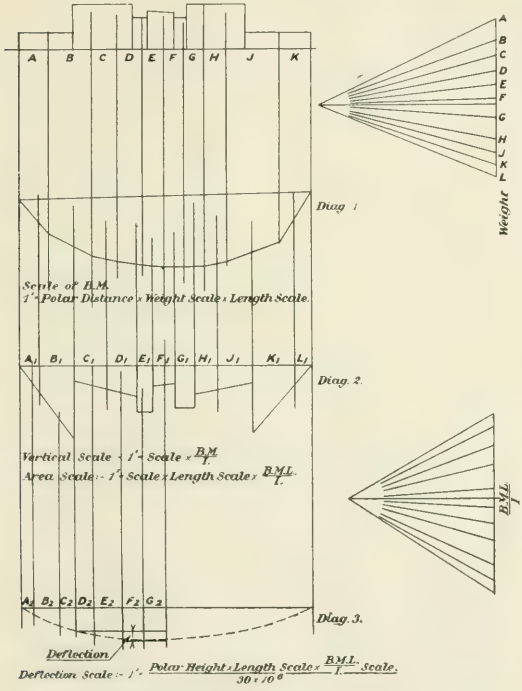
that extreme accuracy of tooth-cutting and maintenance of accurate alignment of bearings be upheld.

It is also advisable that the tooth pressure should be light and uniformly supported across the whole width of the face.

For the rigid type of mounting if p.c.d. = pitch wide diam. the length of tooth face may be taken roughly as equal to

$$\frac{\text{Total pressure on each pinion}}{200 \times \sqrt{\text{p.c.d. pinion}}},$$

or taken so that the maximum pressure per unit width of face does not exceed (in double-reduction gears), 500 lbs. on first-reduction pinion, and 600 lbs. on second reduction. Much higher pressures have been recorded with single-reduction gears, and it should be remembered that the figures given above vary with the diameter of the pinions. The shafts must be of sufficient diameter and rigidity to withstand torsional stresses and bending moments due to loads on the teeth, matters which will be dealt with later.


A B C D E F G H J K
Weight
Diag. 1
Scale of B.M.
1" = Polar Distance × Weight Scale × Length Scale.
A1 B1 C1 D1 E1 F1 G1 H1 J1 K1 L1
Diag. 2.
Vertical Scale = 1" = Scale × B.M./I
Area Scale = 1" = Scale × Length Scale × B.M.I./E
B.M./I
A2 B2 C2 D2 E2 F2 G2
Diag. 3.
Deflection
Deflection Scale = 1" = Polar Height × Length Scale × B.M.I./I.—Scale.
30 × 10⁶

Graphical Determination of Maximum Deflection of Pinion.

The oil lubrication of high-speed gearing must be absolute, copious and fool-proof, as it is most essential to form an oil film between the teeth, and carry away quickly the heat generated by friction.

The oil should be sprayed, under pressure, directly on to the teeth at the point where wheel and pinion teeth come into engagement, because if the wheels merely dip into an oil bath, the friction caused by churning involves a considerable loss of power. A separate system is usually provided for the bearings, and the oil (in both systems) is drained off and cooled in suitable coolers, which are designed on condenser lines. A governor is fitted to cut off steam to the turbines if the oil supply fails. The quantity of oil required varies according to the designer's ideas, and also with the climate in which the ship is intended to run.

(1) For bearings, allowing for an inlet temperature of 100° Fahr. and final temperature of 120° Fahr., the quantity of oil (gallons per square inch surface per minute), $Q = 0{\cdot}00082\ V$, where V = velocity of periphery in feet per second. Therefore, gallons per hour = $Q \times 60 \times$ surface of white metal. The speed of oil in pipes should not exceed about 6 ft. per second.

(2) For Gearing. It is difficult to lay down any rule for this, but the amount of oil for the sprayers may be taken approximately as one jet of $\frac{3}{16}$-in. bore for every 250 h.p. transmitted, as a minimum. With a 1-in. diam. supply pipe, we get :—

$$\text{Gallons per hour} = \frac{\text{Speed of oil in ft./sec.} \times \text{Area of pipe sq. ins.}}{0{\cdot}385}.$$

The oil systems should have a thorough system of mechanical, magnetic and water separating filters, and should be arranged for a copious supply.

Materials.

Pinion-forgings should be of nickel-steel, with a trace of chromium or manganese, in order to possess the necessary tensile strength, combined with ductility and shock resisting properties. A Brinell hardness number of about 220 should be obtained, and the alloy should possess a tensile strength of 40 to 45 tons per square inch, with an elongation of 16 to 20 per cent. The pinions should not be hardened after the gears are cut, otherwise, owing to the long-face width, the resulting distortion will give rise to irregular wear and undue noise. For wheels, mild-steel forgings containing 0·3 per cent. carbon may be used for the shrouds (in which the teeth are cut), which are, in merchant practice, shrunk on to the balanced cast-iron centres. The cast-iron wheels should be made for adequate strength, as the extra weight gives a beneficial fly-wheel effect which tends to damp out any vibrations set up by the propeller, &c.

An empirical rule for calculating the stress (due to torque) in the discs of the wheel, is as follows :—

S.H.P. = Shaft horse-power per wheel at N r.p.m.
R = Radius of wheel to pitch circle (inches).
L = Length of arm from boss (inches).
A = Number of arms per circumference.
n = Number of discs per wheel.

All dimensions in inches. (See diagram on next page.)

$$\text{Stress} = \frac{\text{S.H.P.} \times 33{,}000}{\frac{\text{R}}{12} \times 2\pi \times \text{N}} \times \frac{\text{L}}{\text{A} \times n \times \text{Z}}. \quad (\text{Z} = \text{Modulus of section.})$$

$$= \frac{\text{S.H.P.} \times 63{,}025 \times \text{L.}}{\text{R} \times \text{N} \times \text{Z} \times \text{A} \times n}$$

$$= \frac{\text{S.H.P.} \times \text{L} \times 378{,}150}{\text{R} \times \text{N} \times \text{B}^2 \times \text{T} \times \text{A} \times n} \quad \text{For rectangular section.}$$

This stress, which is for wheels formed of discs with holes and the material between the holes being taken as the arm should not exceed 1,000 lbs. per sq. in. for cast-iron, which provides for a stiff wheel. The diameter of the pinion is limited chiefly by the critical whirling speed of transverse vibrations. These vibrations are due to the sagging of the shaft, due to its weight and the centrifugal force acting on the shaft ; and the speed at which the period of these vibrations coincides with the period of the natural vibrations (or "tone") of the shaft is called the critical speed, and depends upon the dimensions of the shaft and on the distance apart of the bearings, and in no way depends upon the power transmitted. This speed is found by determining the maximum deflection of the pinion, which is usually done graphically, as shown in the opposite column.

First divide the pinion up into any number of convenient parts and ascertain weights of each of these parts. Draw the first polar diagram and produce vertically the lines passing

to centre of gravity of each part and from the polar diagram draw Diagram I, which is the bending moment diagram.

Take B. M. at sections AB, BC, &c., and divide each by the moment of inertia at that section.

From these results draw Diagram II

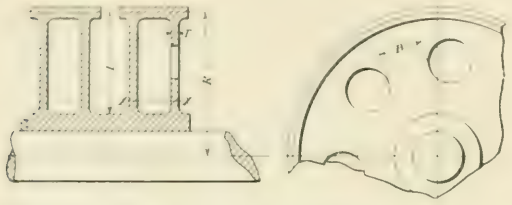
Project the C. of G. of each area, and draw polar diagram for the areas $\frac{B.M.}{I}$.

Then draw Diagram III, which is the deflection diagram. Measure the maximum deflection between the bearings.

When the deflection is determined, a fairly accurate rule is to take—

$$\frac{200}{\sqrt{\text{Deflection in inches}}}$$

as the critical speed; and this must exceed the running speed by not less than 60 per cent.



Sketch showing Symbols used in Empyrical Rule for Calculating Stress.

A more accurate formula for finding the critical speed is due to Baumann, which is :—

$\omega^2 = \frac{(1{\cdot}05)\,g}{\delta}$, where ω = Angular velocity in radians.

δ = Max. deflection in inches.

A still more accurate rule was given in *Engineering* (July 1, 1921), and works out very well in practice :—

$$\omega^2 = 386{\cdot}4 \frac{(\delta_1 + \delta_2 + \delta_3 + \delta_4)}{\delta_1^2 + \delta_2^2 + \delta_3^2 + \delta_4^2}.$$

In this case we assume the pinion to be a cantilever fixed at the aft end only. Draw the curve of total shear. Divide this total shear (S_1) into four equal parts and find the deflections at these points.

Taking an example, where the total shear was 4,076 lbs., we get

at $\frac{S_1}{4}$, deflection $\delta_1 = \frac{1{\cdot}67 \times 6}{10000} = {\cdot}001002.$

$\frac{S_1}{2}$ ... $\delta_2 = \frac{1{\cdot}82 \times 6}{10000} = {\cdot}001092.$

$\frac{3S_1}{4}$ ... $\delta_3 = \frac{1{\cdot}7 \times 6}{10000} = {\cdot}001020.$

$\delta_4 = 1{\cdot}002 \times 10^{-3}$, $(\delta_4)^2 = 1{\cdot}004 \times 10^{-6}$.

$\delta_2 = 1{\cdot}092 \times 10^{-3}$, $(\delta_2)^2 = 1{\cdot}192 \times 10^{-6}$.

$\delta_3 = 1{\cdot}020 \times 10^{-3}$, $(\delta_3)^2 = 1{\cdot}040 \times 10^{-6}$.

$$\omega^2 = 386{\cdot}4 \times \frac{10^{-6}}{10^{-6}} \times \frac{2{\cdot}431}{2{\cdot}491} = 377500.$$

$\omega = 614{\cdot}5.$

$$\text{Critical Revs.} = \frac{614{\cdot}5 \times 60}{2\pi} = 5880 \text{ r.p.m.}$$

By Baumann's formula

$$\omega^2 = \frac{1{\cdot}08 \times 386{\cdot}4}{{\cdot}001105} = 378200.$$

$\omega = 615.$

Critical revolutions = 5,890 r.p.m.

By approximate rule

$$\text{Critical Revs.} = \frac{200}{\sqrt{{\cdot}001105}} = 6020 \text{ r.p.m.}$$

Some firms use a constant of 187, which gives

$$\text{Critical Revs.} = \frac{187}{\sqrt{{\cdot}001105}} = 5630 \text{ r.p.m.}$$

When fixing up the dimensions of gearing, due regard should be given to the following :

$$\text{Circumferential speed in ft. sec.} = \frac{\text{Diam. Wheel} \times \text{Revs.} \times \pi}{720}$$

and should not exceed 140 ft. per second.

$$\text{Torsional load} = \frac{\text{S.H.P.} \times 126000.}{\text{Diam.} \times \text{Revs. Pinion}}$$

and load per unit width of face

$$= \frac{\text{Torsional load}}{\text{Width of Face}},$$

and should conform to limits given earlier in this paper.

Helical gear is necessary for high-speed work, as straight-cut spur gears, owing to unavoidable inaccuracies of cutting, have an irregular tooth action, causing vibration, noise and wear. Moreover, with helical gearing, we get continuity of contact instead of merely an intermittent tooth after tooth contact. The Maag Gear Company, of Munich, however, claim to have brought accuracy of cutting to a very high state of perfection, with the aid of special machines, and advocate straight-cut spur gearing. Nevertheless, the great majority of marine gearing in use is of the double-helical type, in which the end thrusts, due to the spiral angle of the teeth, are balanced. In tooth design, with involute contour teeth, the teeth are cut on the normal so that tooth proportions are calculated from the normal pitch. The line of action is a straight line and lies on a tangent to the two base circles, and passes through the meeting point of the two pitch circles.

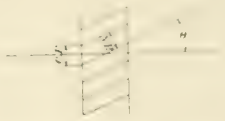


Diagram explaining Terms used in Formula for Number of Teeth.

The pressure angle is the angle formed by the line of action and the common tangent to the two pitch circles, and is taken as 14½ deg. for the standard involute tooth. The spiral angle is now made 30 deg., which gives increased tooth contact and strength over the original 45 deg. angle, which latter was originally adopted to keep the noise at a minimum. The teeth are generated by a Milling process on a "Hobbing" machine.

The hob is a worm, in the thread of which are longitudinal cuts in order to form cutting edges. The wheel to be cut is mounted on a horizontal revolving table and the hob is fed vertically down the face of the blank. The helix angle is formed by setting the hob at an angle and establishing the correct rate of travel of the table and hob-carrier. Double-helical gearing must have a central gap of about 3 in. between the right and left-hand spirals to allow the hobs to run out. Each helix is usually cut separately, and the axis of the hob must be inclined so that its thread on the cutting side is parallel to the spiral angle of the gear.

The following formula can be used for determining the number of teeth on a wheel.

If θ = Spiral angle.

N.P. = Normal pitch. C.P. = Circular pitch.

N.P. = C.P. $\times \cos \theta$.

Number of teeth $= \frac{\pi \times \text{P.C.D.} \times \cos \theta}{\text{N.P.}}$, and must be a whole number.

(To be continued.)

NOTE ON APPROXIMATE INTEGRATION.

By E. V. TELFER, M.Sc.

The subject of approximate integration, lying as it does at the very foundation of naval architecture, has received considerable treatment and attention from both mathematicians and naval architects during the last century. Broadly speaking, the heritage of this endeavour consists in the possession and general application of Simpson's, the trapezoidal or Bezout's, Tchebycheff's and, finally, Gauss's rules. Of these, Simpson's and the trapezoidal are by far the most widely used. The general British opinion is that Simpson's is the more accurate; and that the trapezoidal, although admittedly simple, is not sufficiently accurate for practical purposes. French and American practice, however, seems to favour the trapezoidal rule and, used in a modified form, it may be shown to be quite as accurate as Simpson's. In the establishing of any of these rules, some assumption as to the mathematical nature of ship curves is usually made, a parabola of the second order being the most usual assumption. However, another method which should make a stronger appeal to the practical man, and which was used by the early French investigators, is worthy of re-examination. In Fig. 1 we have a typical curvilinear figure. This figure is divided up in the usual way into a number of equal intervals of base length h. Replacing the curved boundary by the inscribed polygon boundary, we obviously obtain a figure whose area is less than that under the curved boundary. This area is, of course, that obtained by the application of the ordinary trapezoidal rule. If, now, each of the intervals h are subdivided, the dividing lines becoming the sides of the elements of a circumscribing polygon to the original curvilinear figure, the sum of the areas of the elemental trapezoids is obviously greater than the area under the curved boundary. The actual area will be somewhere between those of the inscribed and circumscribed polygons. Thus, if I and E are these respective areas, the actual area under the curve may be expressed as

$$A = I + \kappa (E - I).$$

A consideration of this equality shows that the correct area could be obtained either by making $I = E$, *i.e.*, by making h indefinitely small with a resulting infinite number of ordinates, or, alternatively, with a finite and convenient number of ordinates, by the correct assessment of the factor κ. Increase in the number of ordinates is not desirable from the labour-saving standpoint, and a correct and convenient choice of κ would seem to offer the best line of examination.

Consider Fig. 1 again. The area of the inscribed polygon—only considering for the sake of brevity one half of the figure, the other half being treated symmetrically—is given by

$$I = h\left[\frac{y_o}{2} + y_1 + y_2 + \ldots\ldots\right]$$

similarly, the area of the circumscribed polygon is given by

$$E = h[y_1 + y_2 + y_3 + \ldots\ldots] + \text{Area ABCD} \ldots\ldots$$

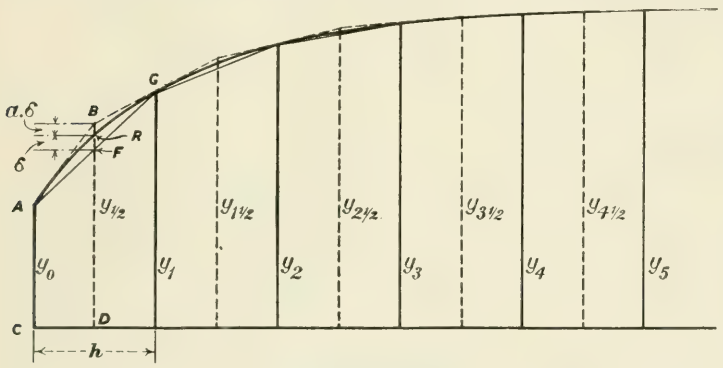


Fig. 1.—Typical Curvilinear Figure showing inscribed and circumscribed Polygons.

To determine the area ABCD some assumption as to the nature of the curve must be made. The only unknown in this trapezoid is the value of BD. From the figure BD is seen to equal $y_{\frac{1}{2}} + a\delta$ where δ is the difference between $y_{\frac{1}{2}}$ and the mean value of y_o and y_1, and a is a factor which depends upon the nature of the curve. The value of a will be investigated later. However, we can now determine BD in terms y_o, $y_{\frac{1}{2}}$, y_1, and a.

Thus—

$$BD = y_{\frac{1}{2}} + a\delta = y_{\frac{1}{2}} + a\left(y_{\frac{1}{2}} - \frac{y_o + y_1}{2}\right),$$

$$= y_{\frac{1}{2}}(1 + a) - \frac{a}{2}(y_o + y_1).$$

From this the area ABCD can be simply determined. Thus—

$$ABCD = \frac{h}{4}\left[y_o + y_{\frac{1}{2}}(1 + a) - \frac{a}{2}(y_o + y_1)\right]$$

$$= \frac{h}{4}\left[y_o\left(1 - \frac{a}{2}\right) + y_{\frac{1}{2}}(1 + a) - \frac{a}{2}y_1\right]$$

$$- h\left[y_o\left(\frac{2 - a}{8}\right) + y_{\frac{1}{2}}\left(\frac{1 + a}{4}\right) - \frac{a}{8}y_1\right].$$

The complete area of the circumscribed polygon may now be written down.

$$E = h\left[y_o\left(\frac{2 - a}{8}\right) + y_{\frac{1}{2}}\left(\frac{1 + a}{4}\right) + y_1\left(\frac{8 - a}{8}\right) + y_2 + y_3 \ldots\right].$$

The difference between the areas of the circumscribed and inscribed polygons thus reduces to—

$$E - I = h\left[-y_o\left(\frac{a + 2}{8}\right) + y_{\frac{1}{2}}\left(\frac{1 + a}{4}\right) - y_1\frac{a}{8} \ldots\ldots\right] \ldots 1.$$

Thus the final area becomes

$$A = I + \kappa h\left[-y_o\left(\frac{a + 2}{8}\right) + y_{\frac{1}{2}}\left(\frac{1 + a}{4}\right) - y_1\frac{a}{8} \ldots\ldots\right] \ldots 2.$$

or, alternatively,

$$A = h\left[y_o\left(\frac{1}{2} - \frac{(a + 2)}{8}\right) + y_{\frac{1}{2}}\frac{\kappa(1 + a)}{4} + y_1\left(1 - \frac{\kappa a}{8}\right) + y_2 \ldots\ldots\right] \ldots\ldots 2A.$$

We must now consider the variation of κ and a with the nature of the particular curves chosen as representative ship curves. Fig. 2 shows the circumscribing and inscribed boundaries on a typical trapezoidal panel, the inscribed boundary

AG being made horizontal for the sake of clearness. Without going into the mathematical geometry of this diagram, it could be simply demonstrated that when the lines AB and BG are tangential to the curve at A and G respectively, and when the curve is respectively a parabola of the first, second, or third order, then κ equals 0, 1 and 2 respectively. With these

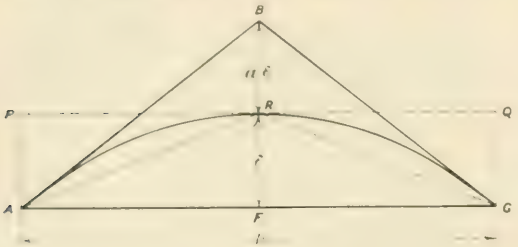


Fig. 2. Circumscribing and Inscribed Boundary on a Typical Trapezoidal Panel.

values of a, definite values of κ are associated. When the curve is a parabola of the first order, *i.e.*, two straight lines, the curve coincides with the circumscribing polygon, and thus κ equals unity. When the curve is a parabola of the second order, the triangle ABG is equal in area to the circumscribing rectangle PAGQ, and thus κ equals $\frac{2}{3}$. When the curve is a parabola of the third order, the triangle ABG is equal to $1\frac{1}{2}$ times the area of the rectangle PAGQ, and as the mean ordinate of a third order curve is three-quarters of that of the circumscribing rectangle, the value of κ will be $\frac{3}{4} \times \frac{2}{3} = \frac{1}{2}$.

Substituting these various values of κ and a into equation 2a we have

3. $a = a$, $\kappa = 1$ $\quad A = h\left|\frac{y_0}{4} + \frac{y_1}{1} + y_1 + y_2 \ldots\ldots\right|$.

4. $a = 1$, $\kappa = \frac{2}{3}$ $\quad A = h\left|\frac{y_0}{4} + \frac{y_1}{3} + \frac{11}{12}y_1 + y_2 \ldots\right|$

5. $a = 2$, $\kappa = \frac{1}{2}$ $\quad A = h\left|\frac{y_0}{4} + \frac{3}{8}y_1 + \frac{7}{8}y_1 + y_2 \ldots\right|$.

Of these expressions number 4 is the best known and is generally referred to as Thomson's rule. Its adoption for displacement calculation work has been recommended, amongst others, by Dr. Hillhouse. Dr. Hillhouse gives an alternative proof in his book on "Ship Stability and Trim." Unfortunately, in spite of the general simplicity of the rule, the factor $\frac{11}{12}$ seems to be a sufficient stumbling block in the way of the rule's general adoption. Expression number 3 is admittedly simple, but, because of its representing a circumscribing polygon, it will yield too high a result. Expression 5, like expression 4, suffers from the complication of the multipliers $\frac{3}{8}$ and $\frac{7}{8}$, and thus has no special attraction which recommends it for practical use. Expression 3 really refers to a special circumscribing polygon ARBG, &c., in Fig. 1, and not to the more general polygon ABG, &c. Considered in this light, expression 3 could be directly deduced. If the particular circumscribing polygon be used in the expression for area,

$$A = I + \kappa(E - I),$$

and, further, if we assume that the actual area under the curve now taken over the full length is the arithmetic mean of those of the circumscribing figures, we can obtain a simple and accurate rule.

Thus—

$$I = h\left(\frac{y_0}{2} + y_1 + y_2 + \ldots\ldots + \frac{y_n}{2}\right)$$

$$E = h\left(\frac{y_0}{4} + \frac{y_1}{4} + y_1 + y_2 \ldots\ldots + \frac{y_{n-1}}{4} + \frac{y_n}{4}\right)$$

$$E + I = A = h\left|\frac{3}{8}y_0 + \frac{1}{8}y_0 + y_1 + y_2 \ldots\right.$$
$$\left. + \frac{1}{8}y_{n-1} + \frac{3}{8}y_n\right| \ldots\ldots 6.$$

This expression may be conveniently termed the mean trapezoidal rule.

If, further, we assume $\kappa = \frac{2}{3}$ in the general expression for area, we can deduce a still simpler and more accurate rule. From the foregoing deduction of expression 6 we have—

$$E - I = h\left(\frac{y_0}{4} - \frac{y_1}{4} + \frac{y_{n-1}}{4} - \frac{y_n}{4}\right)$$

$\therefore$ as

$$A = I + \kappa(E - I),$$

we have

$$A = h\left(\frac{y_0}{2} + y_1 + y_2 + \ldots\frac{y_n}{2}\right) + \frac{2}{3}h\left(\frac{y_0}{4} - \frac{y_1}{4} + \frac{y_{n-1}}{4} - \frac{y_n}{4}\right)$$
$$= h\left(\frac{y_0}{3} + \frac{y_1}{6} + y_1 \ldots\ldots y_{n-1} + \frac{y_{n-1}}{6} + \frac{y_n}{3}\right) \ldots\ldots 7.$$

This expression is practically identical with that known in France as the corrected trapezoidal rule*. In this particular rule the more correct value of $y_{\frac{1}{2}}$, *i.e.*, $y_{\frac{1}{2}} + a\,\delta$, as mentioned earlier in this article, is used, being obtained by actually drawing the tangent to the end ordinate, thus enabling the corrected half-ordinate to be directly measured. For most ship curves the error involved in neglecting the $a\,\delta$ correction is insignificantly small, and the rule given in expression 7 yields a remarkably accurate result. To illustrate the comparative accuracy of the more convenient of the various rules, they are applied in the following table to the determination of the area of a sine curve between the limits of 0 to π.

Station.	Ordinates.	Corrected Trapezoidal.		Mean Trapezoidal.		Thomson.		Simpson's 1st.	
0	0·0000	$\frac{1}{2}$	—	$\frac{1}{2}$	—	$\frac{1}{2}$	—	$\frac{1}{2}$	—
$\frac{1}{2}$	·1564	$\frac{1}{2}$	·026067	$\frac{1}{2}$	·01955	$\frac{1}{2}$	·05213	2	·3128
1	·3090	1	·3090	1	·3090	$1\frac{1}{2}$	·28325	$1\frac{1}{2}$	·4635
2	·5878	1	·5878	1	·5878	1	·5878	4	·23512
3	·8090	1	·8090	1	·8090	1	·8090	2	1·6180
4	·9511	1	·9511	1	·9511	1	·9511	4	3·8044
5	1·0000	1	1·0000	1	1·0000	1	1·0000	2	2·0000
6	·9511	1	·9511	1	·9511	1	·9511	4	3·8044
7	·8090	1	·8090	1	·8090	1	·8090	2	1·6180
8	·5878	1	·5878	1	·5878	1	·5878	4	2·3512
9	·3090	1	·3090	1	·3090	$1\frac{1}{2}$	·28325	$1\frac{1}{2}$	·4635
$9\frac{1}{2}$	·1564	$\frac{1}{2}$	·026067	$\frac{1}{2}$	·01955	$\frac{1}{2}$	·05213	2	·3128
10	·0000	$\frac{1}{2}$	·0000		-	$\frac{1}{2}$	—	$\frac{1}{2}$	—
		Σ =	6·365933		6·35290		6·36656		19·0998
							÷ 3 =		6·3666

* See Lamoruche, "Theorie du Navire" (Challame, Paris, 1921) page 54.

The correct area under this curve is 2·0000, and the common interval being $\frac{\pi}{10}$, we have $\frac{\pi}{10} \times E = 2 \cdot 000$. $\therefore$ E should equal $\frac{\pi}{20} = 6 \cdot 3662$. Thus, the percentage errors in the Corrected Trapezoidal, Mean Trapezoidal, Thompson's and Simpson's rules are respectively —·004 per cent., — ·21 per cent., + ·0056 per cent., + ·0063 per cent. Similarly, if the area had been obtained by the ordinary trapezoidal rule, the error would have been —··82 per cent. Owing to the ordinates of the sine curve only being taken to the fourth decimal place, the quantitative values of the above percentages will not necessarily be reliable. However, they may be taken as illustrating the relative accuracies of the various rules.

Admiral Taylor, in 1895, in a paper before the American Society of Naval Architects, dealt very thoroughly with the question of requisite accuracy in ship displacement calculations. The sources of inaccuracy likely to influence the calculation of displacement are enumerated by Admiral Taylor as follows :—

(*a*) The sections as plotted do not represent exactly the sections of the vessel as built.

(*b*) The ordinates for calculations cannot be measured exactly.

(*c*) The methods by which the volume of the vessel is calculated from the ordinates are necessarily approximate only.

(*d*) The density of the water in which a vessel floats is not accurately known but in exceptional cases.

(*e*) The curve of displacement is necessarily approximate, and we cannot read it exactly.

(*f*) When a ship is afloat, the mean draught can be determined only approximately.

Considering only the external and physical sources of inaccuracy, Admiral Taylor pointed out that these can amount to as much as one-half of one per cent. of the displacement. In the choice, therefore, of any approximate method of integration, such a method should yield no more than this percentage inaccuracy and, preferably, should be even correct to within $\frac{1}{4}$ per cent. of the true displacement. It is further desirable that the rule adopted should slightly underestimate the displacement in order that the inaccuracy may be treated as a margin against possible contingencies. From these considerations Admiral Taylor was lead to adopt the ordinary trapezoidal rule with 21 ordinates for purposes of displacement calculation. Moreover, as an error of $\frac{1}{4}$ per cent. is permissible, a vessel whose mean beam is 40 ft. need only have the ordinates of the waterplane measured to the nearest 0·1 of a foot, and hence the half-ordinates to the nearest 0·05 of a foot. Thus, it is sufficiently accurate in vessels of this size to measure the ordinates to the first decimal place. This fact was very clearly brought out in an example given by Admiral Taylor in the paper previously referred to, for a greater accuracy was actually obtained in the calculation of a mathematically lined vessel whose correct displacement was 10,177·5 tons, by using ordinates correct to one decimal place than by using ordinates correct to two and three decimal places. To obtain the same accuracy as the ordinary trapezoidal, naturally fewer ordinates may be used with the Simpson's, the corrected trapezoidal and the mean trapezoidal rule. The sine curve used in the example on page 8 was integrated by the ordinary, the corrected and the mean trapezoidal rules, over a range of number of ordinates and the percentage errors obtained. These are plotted in Fig. 3 to a base of number of ordinates, and only refer to such curvilinear figures as have their least ordinate at one end and greatest at the other. For figures such as waterplanes, whose least ordinates are at each end and their greatest amidships, the number of ordinates should be halved and then increased by one, in order to obtain the relative accuracy from Fig. 3. Thus the accuracy of 21 ordinates with the ordinary trapezoidal rule in a displacement calculation is obtained from Fig. 3 for 11 ordinates and is $\frac{1}{4}$ per cent. In plotting the accuracy curves for the corrected and mean trapezoidal rule, the end half-ordinates are not included in the number of ordinates in Fig. 3. An examination of this figure shows that

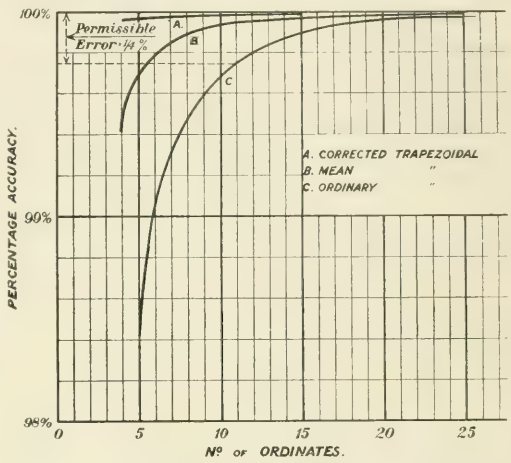


Fig. 3.—Percentage Accuracies with Various Rules.

the percentage accuracy increases at first rapidly with the number of ordinates used and subsequently more slowly. Doyere has shown* that by doubling the number of intervals the percentage error is decreased by more than half. Mr. W. M. Gray, in his recently published work on "Naval Architecture," has shown, in the case of a parabolic segment integrated by the trapezoidal rule, that by doubling the number of intervals the error is decreased to one-quarter. This fact can be quite simply shown to hold good in the general case. From the special circumscribing polygon from which the corrected trapezoidal rule has been deduced we saw that the area could be expressed as $A = I + \kappa h \left(\frac{y_{\frac{1}{2}}}{4} - \frac{y_0}{4}\right) + \kappa h \left(\frac{y_{n-\frac{1}{2}}}{4} - \frac{y_n}{4}\right)$.

Reference to Fig. 4 shows that the expression $h \left(\frac{y_{\frac{1}{2}}}{4} - \frac{y_0}{4}\right)$ is merely the area of the triangle ASR. If now the chord AR and the Arc AR are practically coincident, when the interval h is reduced to $\frac{h}{m}$, the value of RS is reduced to $\frac{RS}{m} = \frac{y_{\frac{1}{2}} - y_0}{m}$. Thus, as for any given size of interval and correct value of κ, the area under the curve is given by the sum of the area of the inscribed polygon and that of the triangle ATS, and its cognate at the other end, it follows that these triangles represent the absolute error in the estimation of area consequent upon the application of the ordinary trapezoidal rule. With the reduced interval $\frac{h}{m}$, the area of the triangle ATS is reduced to $\kappa \left(\frac{y_{\frac{1}{2}} - y_0}{m}\right) \times \frac{h}{4m}$, *i.e.*, in the ratio of $\frac{1}{m^2}$. In the special case of $\frac{1}{m} = \frac{1}{2}$ we see therefore in the case of the ordinary trapezoidal rule, that the error is reduced to one-quarter of its previous value. The same

* See Bourdelle, "Theorie du Navire" (Paris, Octave Doins et Fils, 1912) Part I. page 260.

proposition can be extended to the other rules in which error only occurs because of a faulty estimate of κ. The difference between the correct and the assumed values of κ (*i.e.*, in Fig. 4, TT$_1$) produces a new triangle of error, and from this the proposition established as before.

The foregoing deliberations enable us to examine and distinguish between the practices of various shipbuilders, and hence arrive at the most efficient number of ordinates. With Simpson's rules the number of ordinates in general use vary from 11 to 21. Fig. 3 shows that with 11 ordinates (*i.e.*, 6 in Fig. 3) sufficient accuracy can be obtained with the use of the corrected and mean trapezoidal rules. Of these two rules, the former is the more accurate and simple, and appears especially suitable for the calculation of the waterplane area portion of a displacement sheet. For the calculation of the areas of the transverse sections, owing to the uncertainty of the keel ordinate, there seems to be no good reason for not using the ordinary trapezoidal rule. If the draught is divided into, say, five intervals, the lowest being, as usual, treated as the keel appendage, the area of each transverse section can

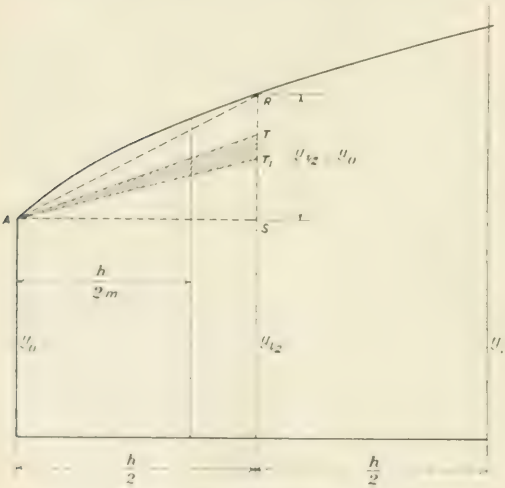


Fig. 4.—Showing Difference between Correct and Assumed Values of "κ."

be obtained by planimeter, and the equivalent keel ordinate for the trapezoidal rule can be calculated and inserted in the displacement sheet. The equivalent keel ordinate device increases the accuracy of the ordinary trapezoidal rule and thus the rule can be safely used in this particular application.

We have now seen that the trapezoidal rule can be applied with all requisite accuracy to the calculation of displacement, and it is interesting to consider the possible saving of effort which its use entails. It is to be regretted that saving of effort has not been more generally recognised in this country from the point of view of a displacement calculation. At the present day the importance of the subject of motion study is being more and more realised in industrial work, and there can be no doubt that it is also important in routine calculation work. While the calculation of displacement is hardly a "bulk production" operation, yet every elimination of waste is always justifiable however small. An examination of an ordinary displacement sheet from the point of view of the possibility of the elimination of redundant mental operations and fatigue will soon repay itself. Standard displacement sheets exist in many yards, and shipbuilders, presumably, rather than interfere with the consistency of their methods choose to perpetuate the mental drudgery of their staff. But apart from this aspect, simpler methods are justifiable commercially and scientifically, no matter what the previous heritage of inertia may be. With the use of Simpson's rule in a displacement calculation, the identity of, say, a waterplane ordinate is completely destroyed. If an error occurs the complete calculation has to be gone over, for a cursory inspection but very rarely reveals the error. With the trapezoidal method the error is very quickly isolated to the class of operation, and from that to the error itself. In 1902, in discussing Mr. Macfarlane Gray's paper on "Simpson's Rules," before the Institution of Naval Architects, Professor Bryan stated:—"the objection to this rule, which suggests itself, is that there appears to be no *prima facie* reason why one set of alternate ordinates should have double the weight of the other set." This argument of Professor Bryan was purely from the scientific and mathematical standpoint, but it serves as a complementary corroboration of the above remarks which have simplicity as their basis. The actual saving of operations in the example given on page 88 is interesting. Considering the multiplication and division of integers as being of equal difficulty (and thus the fractional multiples $\frac{1}{1\frac{1}{2}}$, $\frac{3}{2}$, &c., being double operations), we see that Simpson's rule involves 12 multiplications, Thompson's 6 to the mean corrected trapezoidal, 2 when the end ordinates are zero. Thus, with the latter rules, 83 per cent. of the waterplane multiplying operations are eliminated, and on the whole calculation some 25 per cent. of the total time can be saved. If a "recount" should be necessary the trapezoidal is immensely superior to the Simpson's.

The above considerations show that the trapezoidal rule, suitably modified, has much to commend it for displacement calculations. Its use in this country, in Germany, and even America, is by no means universal. At the present time, when labour-saving ideas are appreciated for their commercial and sometimes their humanitarian value, it is possible that the trapezoidal rule will receive greater recognition. Throughout this note no attempt has been made to deal in detail with the Tchebycheff and Gauss rules. It is doubtful whether the latter are ever used in shipbuilding work. The former seems to be increasingly used in stability calculations in this country, but in "trapezoidal" countries the Tchebycheff has no recommendation for itself. It can be safely said that it is only due to the complication of Simpson's rule that the Tchebycheff has made any headway in this country. With the ten intervals recommended for use with the corrected trapezoidal rule, the work done on the original displacement sheet can be immediately transferred or extended to the construction of Bonjean curves for trim, launching and flooding calculations. They can, of course, be used as a stability body plan with all the advantages from the point of view of integrator manipulation possessed by the Tchebycheff ordinates. The practical man cannot quite see how Tchebycheff's rule accomplishes what it undoubtedly does and chooses, as a result, to adhere to the Simpson. In some yards the Tchebycheff is used as a check upon the Simpson displacement calculation. It is doubtful whether this procedure is justifiable; the better policy would seem to be to eliminate work rather than to duplicate it.

British Exhibition Ship.—The directors of the British World Trade Expeditions have purchased the Orient Liner *Orontes*, which will be known in future as the ss. *British Trade*, and will be refitted by Livingstone & Cooper Limited, of Hessle, Hull.

The Reid-Cooper Pump.—We learn that arrangements have been concluded whereby J. Samuel White & Co. Limited, East Cowes, have secured the exclusive British licence to manufacture and sell the Ried-Cooper Electric Immersible Pump, described in our columns last October. This arrangement took effect at the beginning of the New Year, and all inquiries should be addressed to J. Samuel White & Co. Limited.

THE WELLMAN-BIBBY COMPENSATING SHAFT COUPLING.

Every marine engineer knows by experience the need for a reliable flexible coupling for main and auxiliary drives. The Wellman-Bibby Company developed a coupling to deal with the severe duties of rolling mills and electric winders and they found that its success was so pronounced that they have applied the same principles with equally satisfactory results to practically every kind of direct-connected drive. The coupling is designed to take advantage of the fact that if a strip of material is bent from the straight to a circle, the stress in the material is inversely proportional to the radius of the circle, provided it is not stressed beyond the elastic limit. The flexible element of the coupling is unrestrained from yielding excepting in such a way that its stress is definitely limited to a predetermined figure. The coupling essentially consists of two steel discs having a large number of flared cross grooves in their circumferences. These grooves flare or widen inwardly and the flaring is accurately done to a given radius.

In the grooves is placed a tempered grid spring made out of rectangular-section material. This spring forms the flexible member and transmits the torque between the coupling halves. The rungs of the spring, when no load is transmitted, are straight, but as torque is applied they deflect against the sides of the grooves and the stress is limited by the radius of the sides of the grooves. The diagram on page 92 shows the action of the spring under variations of load right up to the point when the spring bears throughout the length of the groove, and the spring material is almost in shear and exceeds the torsional strength of the shaft. On the outside there is a shell and two junk rings which are held in position by two inverted piston rings sprung into grooves in the shell. The coupling is therefore exceedingly simple and has in fact a less number of parts than ordinary bolted flanged couplings. The spring chamber is filled with grease, or, for very high speeds, it is made oiltight. The coupling is very flexible to allow for want of alignment and resilient to absorb shocks and vibrations so that it provides very good running conditions for any direct-connected drive. A number of the couplings have been fitted with unqualified success to geared-turbine ships, the couplings generally being fitted between the turbine and the gear pinion. It has, however, been lately realised that it is a great advantage to fit the coupling on the low-speed or main propeller shaft, in order

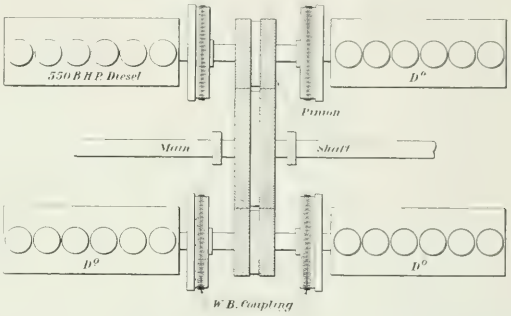
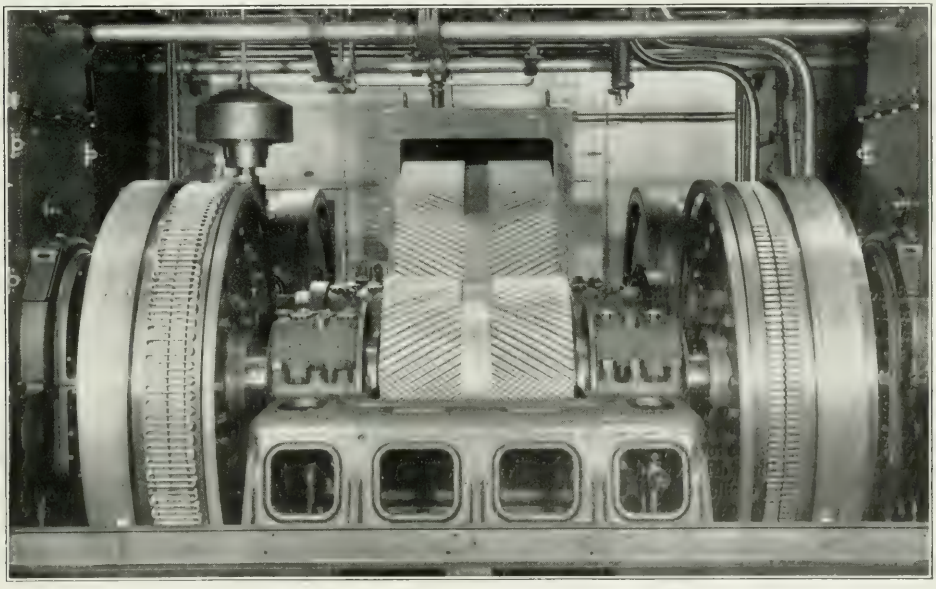


Diagram showing Arrangement for a Geared Diesel Engine Drive.

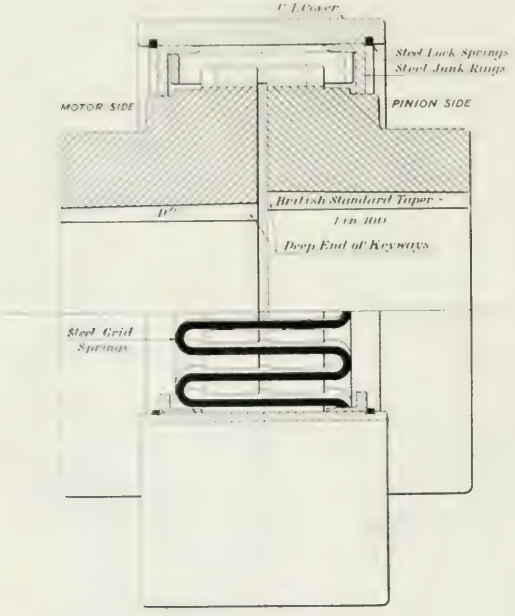


Four Six-Cylinder Diesel Engines coupled to Propeller Shaft with Connections through Bibby Couplings.

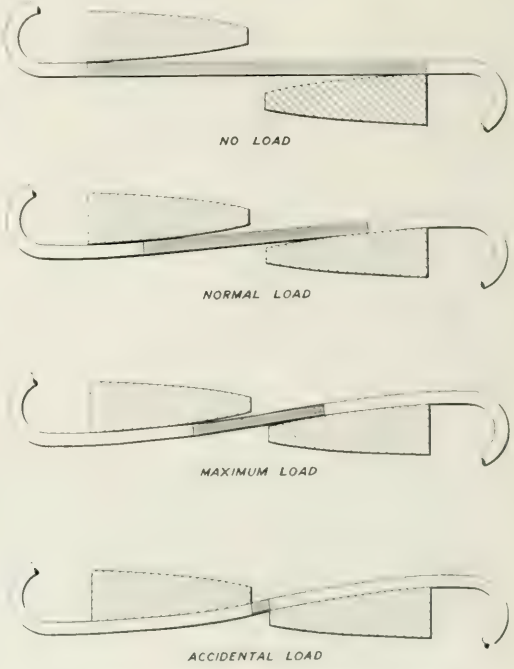
that the severe vibrations from the propeller are not transmitted to the gearing.

One most important application of this coupling has been to Diesel engine driven ships, and the photograph and diagram reproduced herewith shows an interesting example. This plant consists of four six-cylinder Diesel engines of 550 b.h.p. each, and the group is arranged with two engines at either side of the propeller shaft, as will be seen. The two engines on one side are connected through Bibby couplings to a pinion which gears with the large wheel on the propeller shaft, and the two engines on the other side connect in a similar manner to another pinion which gears with the common propeller shaft wheel. One coupling, on the starboard side in the photograph, shows the shell removed and the grid springs in position, the other coupling shows the springs removed, and the two couplings on the port side have the shells in position.

On all Diesel engine drives, and especially where gearing is



Sketch showing Bibby Flexible Coupling.



Action of Spring under Variations of Load.

concerned, account has to be taken of the cyclic speed variation of the crank shaft, and where there is no flywheel this is very considerable. The cyclic speed variation is resisted by the uniform angular velocity of the driven shaft. On tests which have been made it has been found that the forward torque during the acceleration periods may be four times the mean torque owing to the resistance which the driven shaft offers to the acceleration. Again, the resistance of the driven shaft to retardation causes a reverse torque of about three times the mean forward torque. On a drive of this kind we therefore have very rapid torsional oscillations, which, if not damped, are apt to prove destructive to the gearing and fatiguing to the shafts. The principle of the coupling is such that, at normal torque, there is a long span of unsupported spring between the coupling halves, and it is then very flexible and resilient to absorb the above-mentioned peak torques without increasing the stress in the spring material.

It should be noted that the couplings on the drive shown in the photograph are about twice the size of those required because they had to be designed to replace couplings of the pin and rubber bush type. Ordinarily the couplings would have been about the same size as the pinion and less in diameter than an ordinary solid flanged coupling. Extensive trials have been made on the couplings on geared Diesel engines, and it has been found that they practically eliminate the destructive effect on gears of vibrations or torsional reciprocations from the prime mover. It will be seen that the couplings occupy very little space in the direction of the axis of the shaft and no case has yet occurred where it has not been possible to replace either a solid or other type of coupling without alterations to the structure of other parts. Reverting to geared-turbine drives, it has been well established that the inevitable torsional oscillations from the propeller are an important cause of the destruction of the gears, and it only remains to be seen how far a Bibby coupling when fixed on the propeller shaft will insulate the gears from the propeller vibrations.

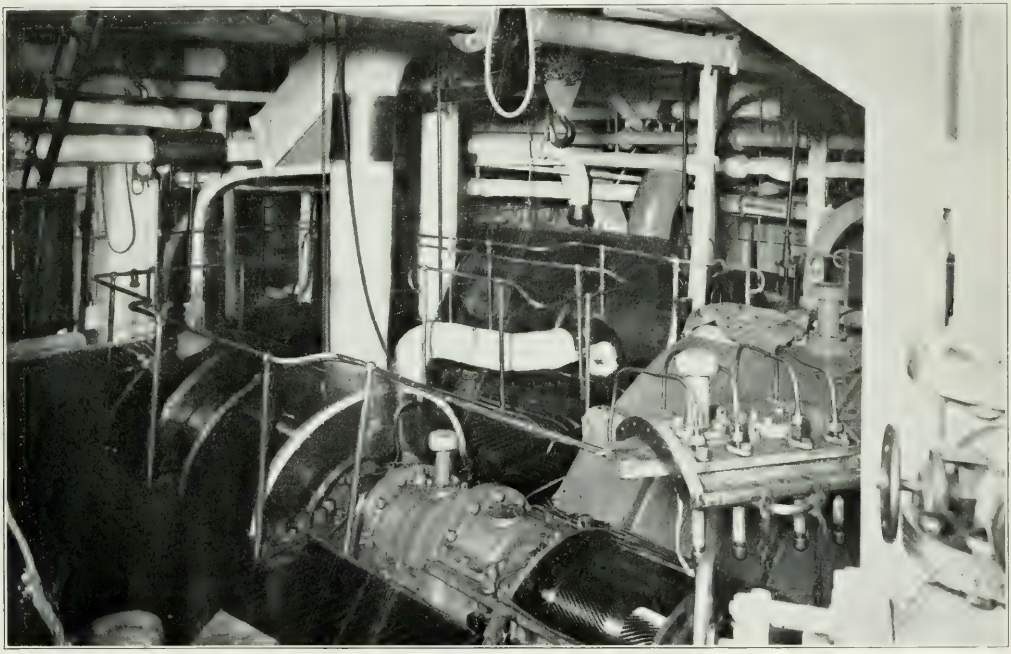
The couplings have already been developed for great capacitites, and several have now been supplied to transmit up to 500,000 lb. ft. torque and there is no feature in them which limits their being constructed of any capacity.

The makers have also produced a line of model couplings which show the action of the steel grid spring in the flared grooves of the coupling halves, and we understand that one of these models will be sent to engineers interested. It is interesting to note that the Falk Corporation, of Milwaukee, has taken out a licence for the United States, and the Citroen Gear Company for France.

MODERN MACHINERY SPACES—II.

The Cunard Company's Cabin-Class Liner "Antonia," built by Vickers Limited, Barrow-in-Furness.

The *Antonia*, whose engine room we are describing this month, is one of the cabin-class vessels built under the Cunard Company's post war scheme. She was constructed by Vickers Limited, Barrow-in-Furness, for the Cunard Canadian service, and sailed from Southampton on her maiden trip on June 15, 1922, bound for Quebec and Montreal.



General View of Starboard Side Turbines and Engine Room, SS. "Antonia," taken from Switch-board Platform.

The following are her leading particulars :—

Length, extreme	536 ft. 0 in.
Breadth, ,,	65 ft. 2 in.
Depth, moulded, to C. deck	43 ft. 0 in.
,, ,, B. ,,	51 ft. 3 in.
Draught	29 ft. 5¼ in.
Displacement	20,770 tons.
Gross tonnage	13,867 tons.
Passengers, cabin	484
,, 3rd class	1,222
Crew	271
H.P. of propelling machinery	8,500 s.h.p.
Speed	15 knots.

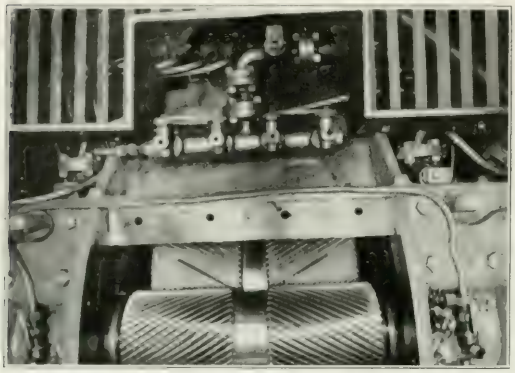
The vessel is classed 100 A.1. at Lloyd's, shelter deck with freeboard, special survey, and has passed the survey of the British Board of Trade and American authorities as a passenger steamer.

The main propelling machinery consists of twin sets of Parsons turbines, driving screws through helical double-reduction gearing. The turbine arrangement is of the two-stage type,the two high-pressure ahead turbines being on the inside, and exhausting into the low-pressure turbines, which are arranged on the outside of the units. Astern turbines are incorporated in the casings of the low-pressure ahead turbines, and the installation develops, when running astern, about 63 per cent. of the ahead power. We illustrate below a comprehensive view of the engine room, taken from the switch-board platform, situated at the aft end, of the machinery space, showing the main turbines and part of the gearing with cover removed, and we also show a close up view of the gearing of the starboard H.P. pinion and intermediate gear wheel. The manœuvring platform is situated on the engine-room floor immediately abaft the boiler-room bulkhead, and a photograph, which we reproduce elsewhere, shows the controls for the starboard unit. The gauges, showing steam pressures to the various units, together with a clock, barometer, and rolling indicator are fitted to a board between the two control units. Engine and boiler room are divided by a watertight bulkhead and can be separated the one from the other by means of a Stone watertight door, the pumps actuating the hydraulic control for this and for the other similar doors being situated close up against the boiler-room bulkhead on the port side, near the Clarke-Chapman forced lubrication pumps. The complete turbine

installation is capable of developing a collective shaft horse-power of 8,500, with propeller revolutions at about 90.

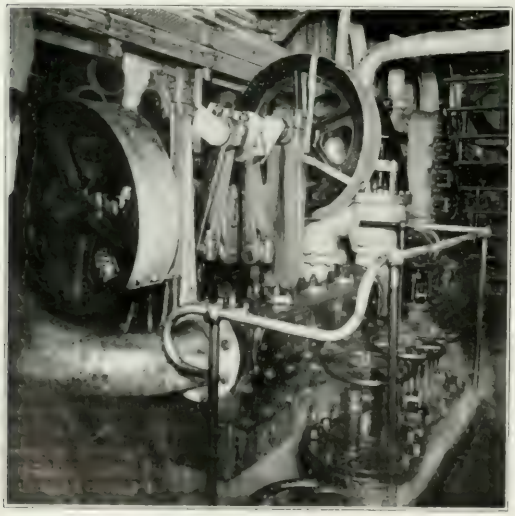
Turbine Details.

The casings of the high-pressure ahead turbine and the steam chest portion of the high-pressure astern turbines are made of cast-steel to withstand the effects of the superheated steam. The high-pressure ahead turbines consist of two rows



View of H.P. Pinion and Gear-wheel, Starboard Side Turbines.

of impulse wheels and five stages of reaction blading, each stage consisting of six rows. The low-pressure ahead turbines are all of the reaction type, and comprise three rows of impulse blading and five stages of reaction blading. Adjusting blocks of the Michell type are fitted to the turbines. The gearing

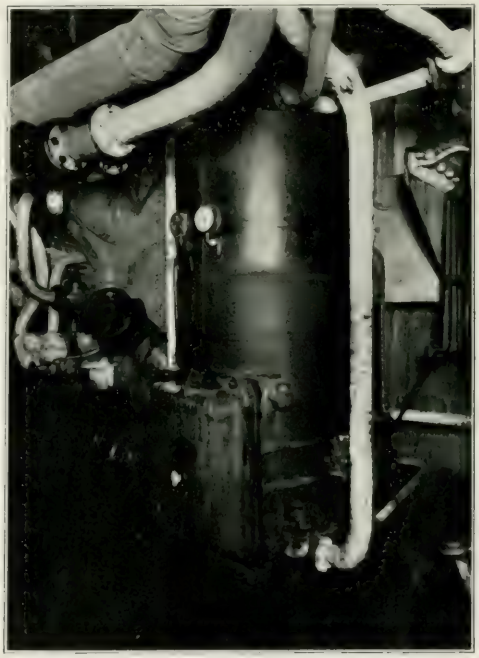


Emergency Motor-Driven Ballast Pump, Port Side.

is of the double-helical type, the main gear-wheel being constructed with a cast-iron centre portion on which a heavy forged steel rim is shrunk on. The H.P. and L.P. pinions are of forged nickel steel. The shaft bearings for the turbines and the gearing are arranged to work under forced lubrication. Thrust blocks of the Michell type are fitted on the main shafts next to gearing to take up the propeller thrust. The main condensers are of the Contraflo type, under-slung on the low-pressure turbine so that the low-pressure turbine is always free from drainage water. The vacuum system embodies all recent improvements and is of the Contraflo system, comprising steam ejectors and Vickers' kinetic reciprocating air pumps, arranged for economical and efficient operation in conjunction with the feed-heating system. The auxiliary condensers are also of the Contraflo vacuum type, and the auxiliary feed system is similar to the main system. The propellers are of the built type, with four manganese bronze blades to each, the bosses being made of cast-steel. All the main shafting is made of hydraulic forged ingot steel, finished bright.

Auxiliary Machinery.

The main circulating pumps, made by W. H. Allen, Son & Co.



View showing one of the Quiggin's Evaporators and Auxiliary Circulating Pumps.

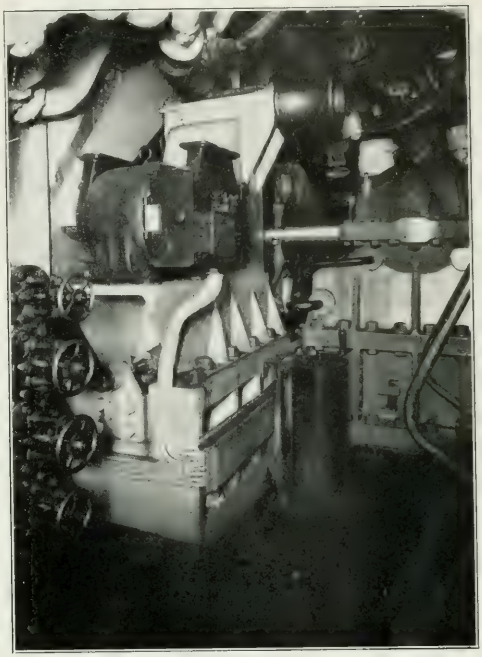
Ltd., Bedford, are of the centrifugal type, each driven by a radial-flow impulse turbine developing 75 b.h.p. at 1,000 r.p.m. The circulating pump for the auxiliary condenser, fitted on the starboard side of the engine room between the evaporators and the main circulating pumps, is also of the centrifugal type, driven by a single cylinder high-speed engine of forced lubrication type.

The boiler feed arrangements are interesting and are illustrated elsewhere. They consist of one turbo-pump of Weir's make, and two of the direct-acting type; the two direct-acting pumps together, and the turbo pumps alone, are capable of supplying the boilers at full power.

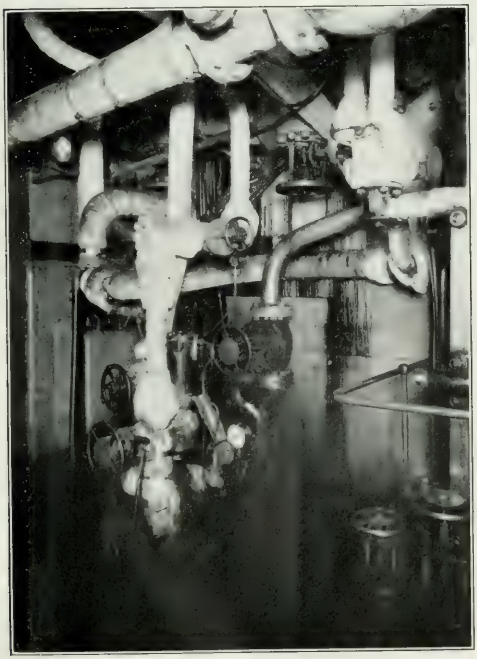
The evaporating plant is illustrated above, it consists of two 50-ton evaporators, of Quiggin's type, made by the Liverpool Engineering Company, fitted with automatic feed and brine ejector. Evaporators are arranged for working singly or in series. There is also one distiller of the

MACHINERY OF CUNARD SS. "ANTONIA."

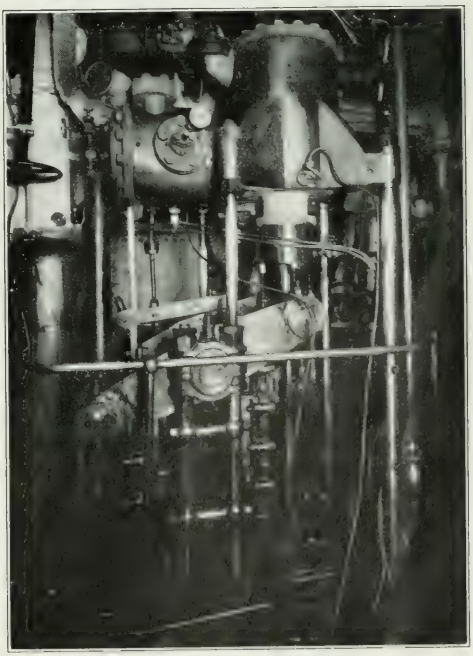
Air Pump for Auxiliary Condenser. Pump Working.



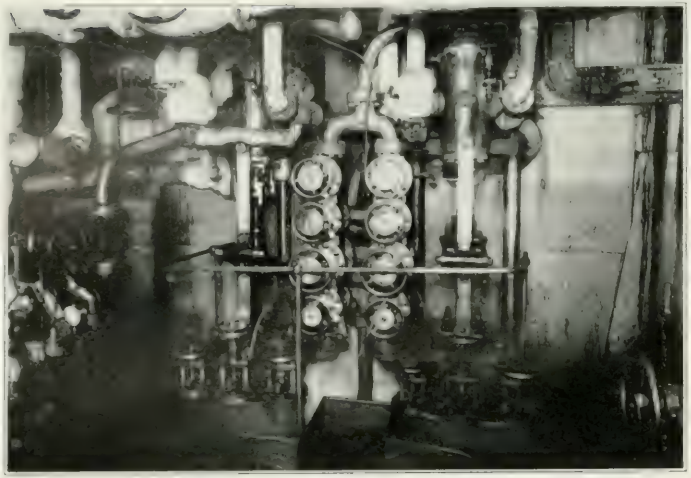
Turning Gear for Starboard Propeller Shaft.



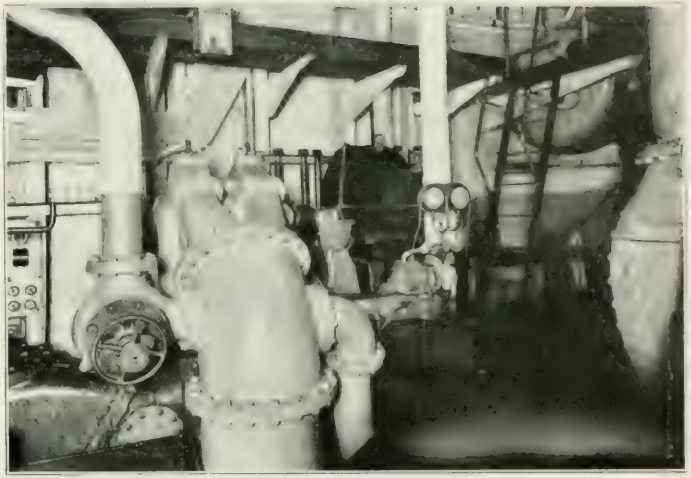
Turbo-Driven Boiler Feed Pump.



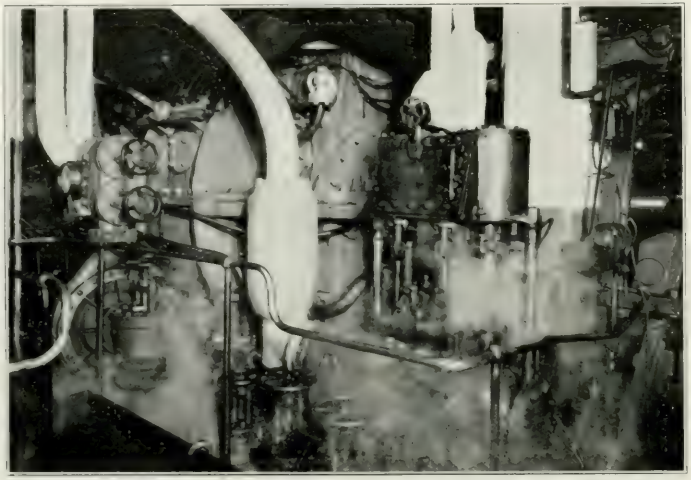
Port Side Main Air Pump.



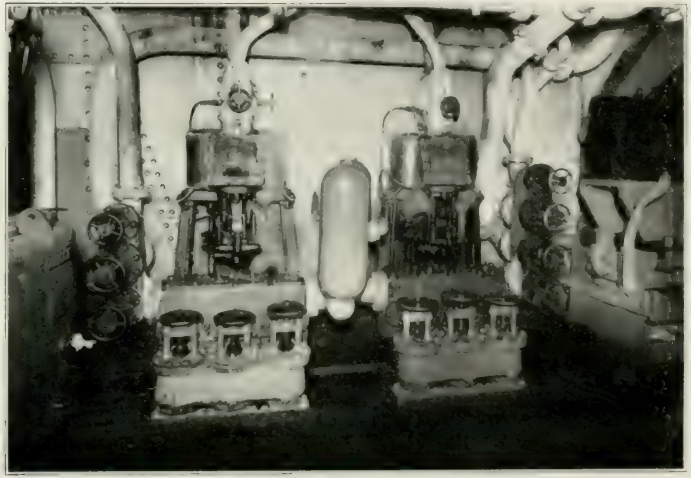
Direct-Acting Boiler Feed Pumps, showing Valve Connections.



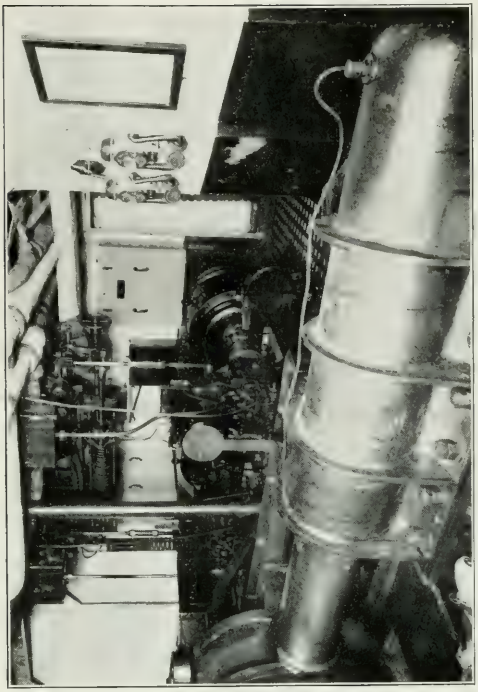
Turbo-Driven Main Circulating Pump for Starboard Turbines.



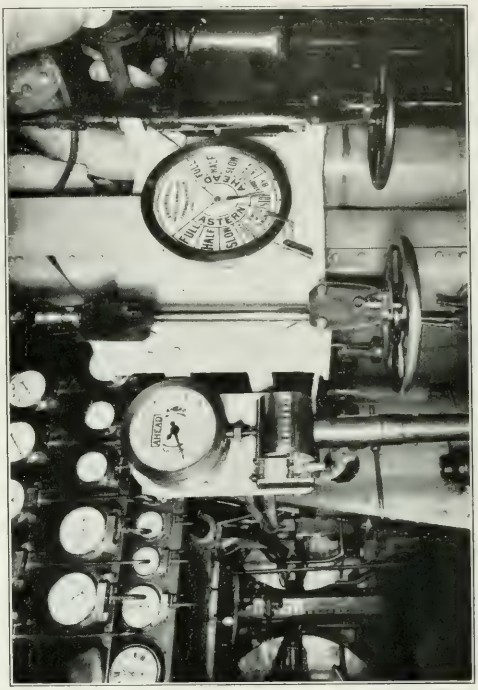
Engine Room, Port Side, showing Motor-driven Sanitary Pump, and Bilge Pump of Cunard Company's Make.



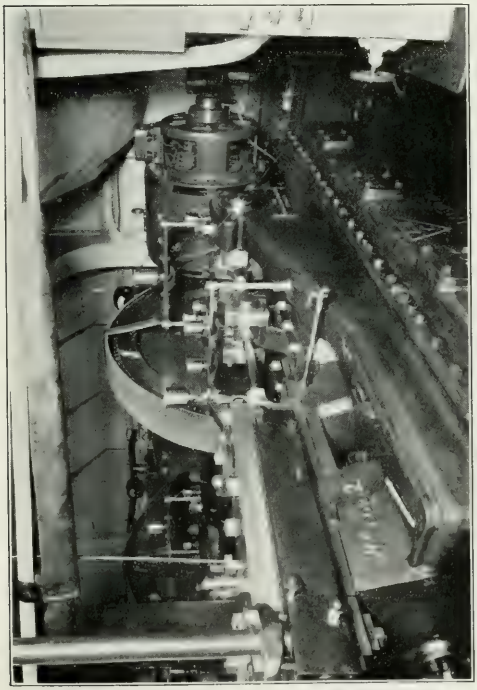
Fresh-Water and Washing-Water Pumps at Aft End of Engine Room.



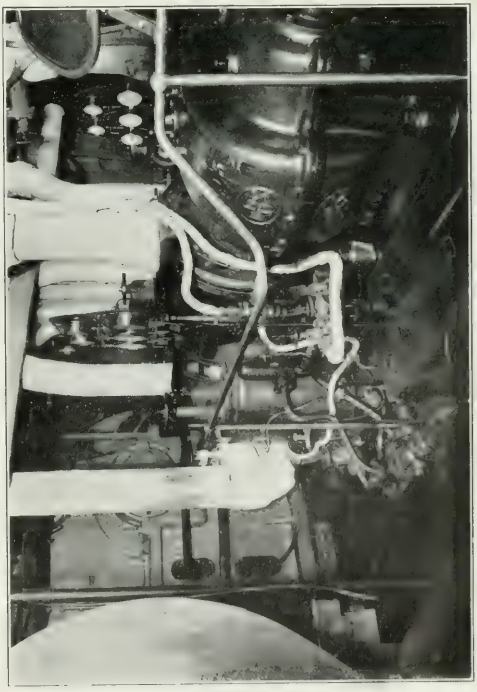
General View of Hele-Shaw Electro-Hydraulic Steering Apparatus.



Control Platform for Port Turbines, showing Telegraphs, Revolution Indicators, &c.

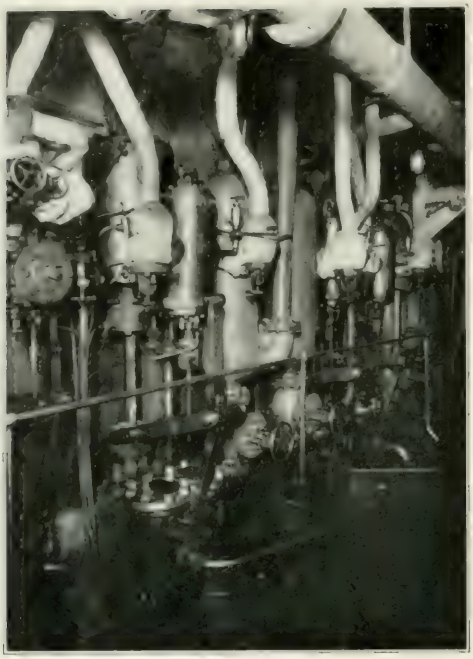


One of the Motor-driven Refrigerating Machines.

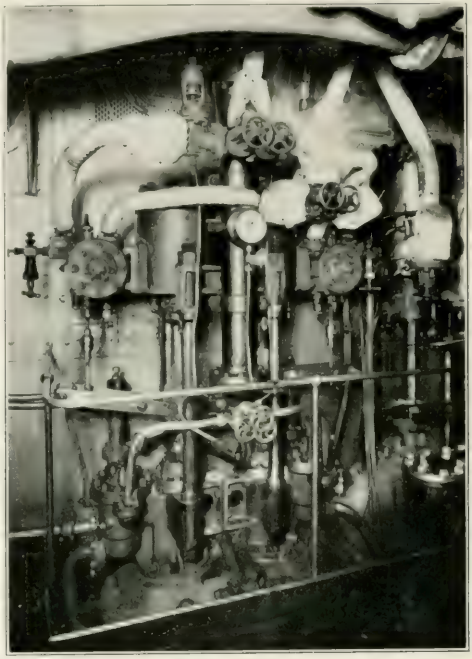


View of One of the Turbo-Generators, showing Turbine and Gear Case.

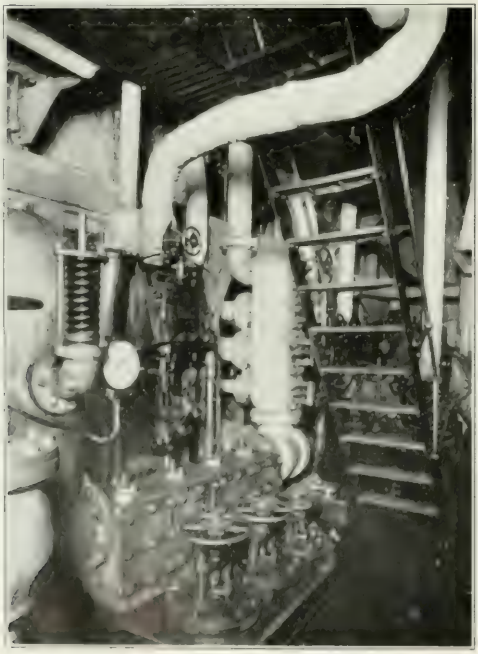
MACHINERY OF THE CUNARD SS. "ANTONIA."



Forced Lubrication Pumps on Boiler Room Bulkhead.



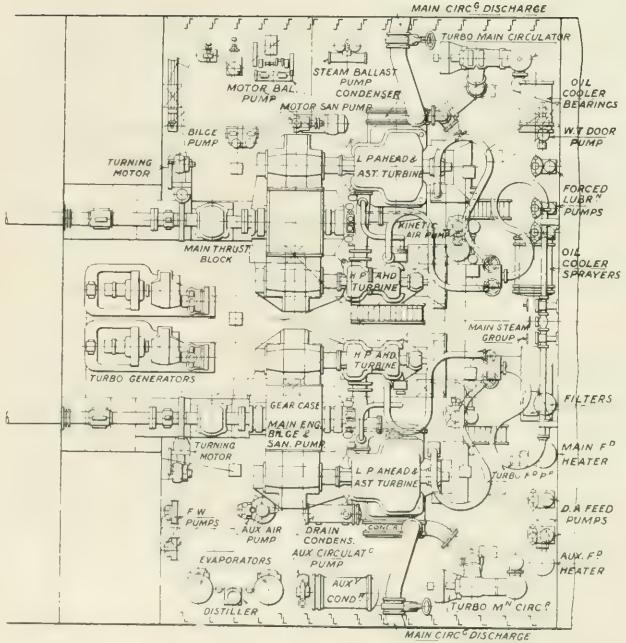
"Stone" Patent Pump, for Operating Hydraulic W.T. Doors.



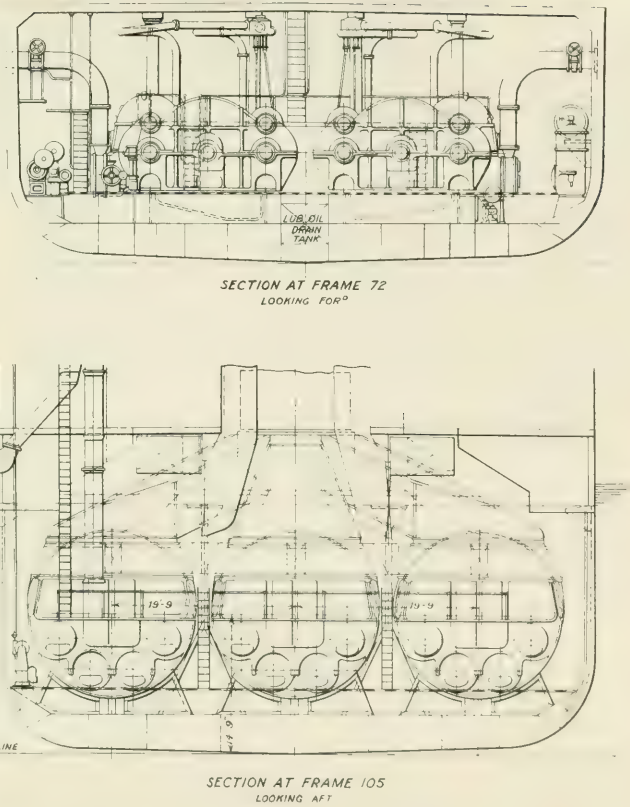
Direct-Acting Ballast Pump and Connections.

Steam-Driven Centrifugal Circulating Pump, for Auxiliary Condenser.

THE MACHINERY SPACE OF THE CUNARD LINER "ANTONIA."



Plan showing General Arrangement of Engine Room.



Sections through Engine and Boiler Rooms.

size type, and this is capable of distilling 50 tons of water per twenty-four hours.

The ballast pumping arrangement is a duplicate one.

There is one pump of the duplex type, made by Dawson & Downie Limited, which has brass pump ends, and also one ballast pump, motor-driven, by Thom, Lamont & Co. Both of these are fitted to draw from the ballast tanks, bilges and sea, and a connection is also provided which permits of the ballast pumps circulating water through the main condensers.

There are two bilge pumps, one of which is illustrated on page 96; this is of the ordinary direct-acting type, and is of the Cunard Company's own make. The other, an emergency centrifugal pump, of Dawson & Downie's make, is driven by an electric motor. The direct-acting pump has gun-metal water ends.

There is one washing water pump of the duplex type, by Thom, Lamont & Co., having brass pump ends, for supplying condensed water for washing purposes, and one fresh-water pump by the same makers for supplying the ship's drinking fresh-water service; this latter also has brass pump ends, and the two pumps which are arranged at the aft end of the engine room are illustrated elsewhere.

There are two fresh-water filters of the Quiggins type. These are fitted with connections from the discharge of the fresh-water and washing pumps.

One sanitary or wash-deck pump, and one bilge pump are arranged at the forward end of the gear case on each main engine. There is also one sanitary and wash-service pump by the Pulsometer Company, electrically-driven, arranged to draw from the sea and discharge through the sanitary and water-service mains.

The forced lubrication system for the turbines is controlled by two pumps of the direct-acting type, by Clarke-Chapman; these pumps are arranged on the port side of the entrance to the boiler room on the boiler-room bulkhead, and are alongside the Stone pump which controls the hydraulic mechanism for the watertight doors. The main air pumps are of the Vickers kinetic type, and are arranged one on each side of the manœuvring platform, as has been mentioned already.

There is a turning gear fitted on each shaft, fitted with a winch barrel and operated by an electric motor, supplied by Electromotors Limited, Manchester.

The auxiliary equipment is completed by :—

One oil-fuel transfer pump, of Clarke-Chapman's make, which draws from the double-bottom tanks and discharges to the oil-fuel settling tanks; one oil-bilge well pump, of Clarke-chapman's make, for pumping out the oil bilges; two Royce Limited electric ash hoists, and a drain condensing plant consisting of condenser, circulating and air pump, combined, by Haywood, Tyler & Co.

Boiler Installation.

The steam-generating installation consists of two double-ended boilers and two single-ended boilers, arranged in one compartment, and constructed to Board of Trade and Lloyd's requirements for a working pressure of 220 lb. per sq. in. The boilers are of the multitubular return-tube type, fitted with Robinson's superheaters, and working with Howden's forced draught. The mean diameter is 17 ft. 6 in., and the mean length of the double-ended boilers is 22 ft. 6 in., and that of the single-ended boilers is 11 ft. 6 in. Each double-ended boiler is fitted with eight furnaces of the Morison suspension type, and eight separate combustion chambers. The single-ended boilers having four furnaces and four separate combustion chambers. The combined heating surface for the two double-ended and two single-ended boilers is 19,442 sq. ft., and the combined grate area 497 sq. ft. The boilers are fitted with smoke-tube type of superheaters, the temperature of the superheated steam entering the turbines being 200 deg. Fahr. in excess of the saturated steam temperature corresponding to the pressure. As has been mentioned, the boilers are arranged to work under Howden's forced draught system, and two sets of single-inlet fans, electrically-driven, are fitted, one motor being arranged to drive two fans. The boiler fronts are arranged for quick conversion from coal to oil burning or *vice versâ* on the White low-pressure system, and the Cumberland electrolytic system for the prevention of corrosion is fitted to all boilers.

Generating Sets, &c.

For generating power for the electric auxiliaries, steering gear, &c., there are two turbo-generators fitted at the aft end of the engine room. Each turbine runs normally at 5,000 r.p.m., and drives generator through reduction-gearing supplied by David Brown & Co., Huddersfield, so that it runs at 1,000 r.p.m. The generators have each an output of 300 kw. at 225 volts and 1330 amps., and were supplied by the English Electric Company, Bradford.

The refrigerating apparatus is motor-driven, and of the type made by the Liverpool Refrigerating Company, the motors being supplied by Newton Bros., Derby. The brine distribution pumps of the direct-acting type, are by Mumfords, of Colchester.

An emergency oil-driven motor generator, situated in a house on the boat deck, supplies current for the steering gear, the wireless plant and the lighting of the vessel in the event of the main generators getting out of action.

We wish to express our indebtedness to the Cunard Company for the facilities placed at our disposal for photographing the machinery of the *Antonia*, and to Mr. Austin, Superintending Engineer of the Line; Mr. Bell, Chief Engineer of the *Antonia*, and others for their assistance.

ENGAGEMENTS FOR MARCH.

Institute of Marine Engineers.—Meetings at the Institute the Minories, Tower Hill, London, E., on Tuesday, March 13, at 6.30 p.m. "The Development of the Air Pump for High Vacuum," by W. A. Dexter. Tuesday, March 20, "Operation of Marine Oil Engines Cylinders and Pistons," by J. Lamb (Member).

Institute of Metals. Annual general meeting at the Institution of Mechanical Engineers, Storey's Gate, Westminster, S.W., on Wednesday and Thursday, March 7 and 8, at 10 a.m. each day. Annual Dinner at Trocadero Restaurant, Piccadilly Circus, London, W., on Wednesday, March 7, at 7.30 for 8 p.m.

Institute of Transport.—Meeting in Lecture Theatre, Institution of Electrical Engineers, Savoy Place, Victoria Embankment, London, W.C., on Monday, March 5, at 5.30 p.m. "Operation of a Railway-owned Port," by Major G. S. Szlumper (Member).

Institution of Engineers and Shipbuilders in Scotland.—Meeting in Rankine Hall, 39, Elmbank Crescent, Glasgow, on Tuesday, March 13, at 7.30 p.m.

Liverpool Engineering Society.—Meetings at the Royal Institution, Colquitt Street, on Wednesday, March 7, at 8 p.m. "Hardening of Steel," by Prof. C. O. Bannister, Assoc.R.S.M., F.I.C. Wednesday, March 21, "Standardisation of Ship Fittings," by C. Le Maistre, C.B.E.

North-East Coast Institution of Engineers and Shipbuilders.—Meeting in Lecture Theatre of the Literary and Philosophical Society, Westgate Road, Newcastle-on-Tyne, on Friday, March 16, at 7.30 p.m.. "Notes on the Economical Performance of Turbine Installations in Merchant Ships," by T. G. Potts (Member).

Manchester Association of Engineers. Annual general meeting in Memorial Hall, Albert Square, Manchester, on Friday, March 23, at 7 p.m.: (1) Election of officers and members; (2) "Suction Gas Producers: Their Construction and Working," by W. Greenwood (Member). *Student Members' Section.* Annual general meeting in Chemical Lecture Theatre, Municipal College of Technology, Sackville Street, Manchester, on Friday, March 16, at 7.30 p.m. Lecture: "Commercial Engineering," by H. Allcock, M.I.Mech.E., M.I.E.E., A.Amer.I.E.E.

A REVIEW OF DIFFERENT TYPES OF MARINE INTERNAL-COMBUSTION ENGINES—PART II.

The Doxford Opposed Piston Engine—The Neptune Engine—The Scott-Still Engine—The Vickers Internal-Combustion Engine—The Werkspoor Engine.

We continue, from page 69, the symposium on different types of marine internal-combustion engines, which was read recently before the North-East Coast Institution of Engineers and Shipbuilders. Part I dealt with the Armstrong-Sulzer, Camellaird-Fullagar and Doxford Opposed Piston Engines. This part concludes the Doxford engine and deals with the Neptune, Scott-Still, Vickers and Werkspoor Engines.

Our description of the Doxford engine ended in the last issue with a general description of the proposed machinery. From this point the builders' statement went on to give the following replies to the *questionnaire* sent out by the Institution, details of which were given on page 63 last month :—

1. *Total brake horse-power of the propelling machinery.*

For 10¾ knots, 1,850 s.h.p. at 96 revolutions.

2. *Overload obtainable in brake horse-power above trial conditions* :—

11½ knots, 2,270 s.h.p. at 103 revolutions.

3. *Mechanical efficiency of the engine, stating the basis upon which the figure has been arrived at.*

Eighty-two per cent. based upon test bed records with four cylinders, 3,000 i.h.p. marine engines, which have given 85 to 86 per cent. mechanical efficiency when working against Heenan & Froude water brake.

4. *Consumption of fuel per brake horse-power when the ship is steaming at the designed speed.*

Diesel Oil .... 0·4 lb. b.h.p. hour	(*a*)	8 tons per day for engine only.
	(*b*)	8½ tons for engine and electric auxiliaries.
	(*c*)	9 tons engine and boiler at sea.
10¾ knots. . . Boiler Oil .... 0·42 lb. b.h.p. hour	(*a*)	8½ tons per day for engine only.
	(*b*)	8¾ tons for engine and electric auxiliaries.
	(*c*)	9½ tons engine and boiler at sea.
Diesel oil .... 0·4 lb. b.h.p. hour	(*d*)	9¾ tons per day engine only.
	(*e*)	10 tons per day engine and electric auxiliaries.
	(*f*)	10¾ tons engine and boiler at sea.
11½ knots .... Boiler oil .... 0·42 lb. b.h.p. hour	(*d*)	10½ tons per day engine only.
	(*e*)	10¾ tons per day engine and electric auxiliaries.
	(*f*)	11¼ tons per day engine and boiler at sea.

5. *The brand of fuel assumed and its calorific value. What fuel would be recommended for continuous running, and whether the engines would run satisfactorily on the heavier boiler fuel oils for a continuous period.*

Diesel oil, 0·895 s.g.—19,200 B.Th.U. lb. higher value ; 18,000 B.Th.U., lower value.

Boiler oil, 0·945 s.g.—18,300 B.Th.U. lb. higher value ; 17,200 B.Th.U. lb. lower value.

We recommend the use of Diesel oils as these have generally a lower percentage of ash, and consequently the wear on cylinder liners and piston rings is reasonable. Boiler fuel oils can be used satisfactorily and continuously and without any lighter fuel oils as a standby, but extra wear on cylinder liners and piston rings would have to be taken into the bargain on account of the hard particles of ash which some of the heavier boiler oils contain.

6. *Total weight of the main propelling engines and auxiliaries in engine room.*

Four hundred and thirty tons with steam auxiliaries and steam up. Four hundred and fifty tons with electrical auxiliaries ready for sea.

7. *Cost per ton-mile in pence and* 6,570 *tons d.w.*

Diesel oil at 87s. 6d. per ton.

Boiler oil at 70s. per ton.

Lubricating oil at 3s. per gallon.

	Steam Auxiliaries. Diesel Oil.	Steam Auxiliaries. Boiler Oil.	Electrical Auxiliaries. Diesel Oil.	Electrical Auxiliaries. Boiler Oil.
10¾ Knot Voyage—				
8½ tons fuel per day			0·0051	
8¾ ,, ,,				0·0043
9 ,, ,,	0·0055			
9½ ,, ,,	....	0·00472	....	
11 gallons per day lubrication	0·0002	same.	same.	same.
11½ Knot Voyage—				
10 tons fuel per day	....		0·0058	
10¾ ,, ,,	0·0062			
10¾ ,, ,,				0·005
11¼ ,, ,,		0·00525	....	
13 gallons per day lubrication	0·0002	same.	same.	same.

8. *Engine-room crew.* 3 Engineers. 3 Greasers.

Sizes and Capacities of Pumps and Auxiliaries in Engine Room.

Lever-Driven Pumps on Main Engine—	
1 Sea-water circulating pump	100 tons per hour.
1 Jacket-water circulating pump	70 ,, ,,
1 Forced-lubrication pump	25 ,, ,,
2 Bilge pumps, each . . .	95 ,, ,,
Auxiliaries.—Steam-driven.—	
1 Ballast pump . .	200 tons per hour.
1 Jacket-water circulating pump	70 ,, ,,
1 Forced-lubrication pump	25 ,, ,,
1 General-service pump	25 ,, ,,
2 Boiler-feed pumps, each	7 ,, ,,
1 Oil-fuel transfer pump	10 ,, ,,
1 Air compressor	120 cubic feet.
1 ,, ...	50 ,,
1 Electric-lighting set .	8 kw.
1 Auxiliary high-pressure fuel pump driven by	7″ · 6″ steam engine.

The Neptune Marine Oil Engine.

By Swan, Hunter & Wigham Richardson, Limited.

The engine proposed will fulfil the following conditions :—

Brake horse-power.—Will maintain 1,840 b.h.p. at 90 r.p.m. during a six hours' trial.

Overload.—At least 10 per cent. would be obtainable above the trial brake horse-power, but 25 per cent. has been obtained on a bench test.

Mechanical efficiency.—Will not be less than 73 per cent. (deduced from bench tests of previous engines).

Fuel oil consumption.—The consumption of fuel oil at the designed service power will be :—

(*a*) 0·43 lb./b.h.p./hour, main engines only.

(*b*) 0·48 lb./b.h.p./hour for all purposes with steam-driven auxiliaries.

(*c*) 0·45 lb./b.h.p./hour for all purposes with electrically-driven auxiliaries.

The usual 5 per cent. should be added to the above figures for guarantee purposes.

Fuel oil.—The above figures are based on "Diesel Oil" with a net calorific value of not less than 18,500 B.Th.U.

The engine will run satisfactorily and continuously on boiler fuel oil reasonably free from ash, the only appreciable effect being a greater accumulation of deposit in the cylinders, &c., involving more frequent opening up for cleaning. The factor having most influence on the deposit appears to be the percentage of hard asphalt, and oils containing high percentages of this should be avoided.

Total weight of machinery installation—430 tons.

Outline Description of Engines and Auxiliaries.

Main engines.—Neptune Marine Oil Engine of the reversible two-cycle single-acting, part scavenging type, having four cylinders 26 in. diameter by 52 in. stroke.

Main compressor.—Of the three-stage type, fitted at the forward end of the engine and driven from an extension of the crankshaft.

Pumps.—The following pumps are arranged on the back of the engine and driven by levers from the piston crossheads :—

2 double-acting scavenge air pumps.
2 jacket water circulating pumps.
1 piston water circulating pump.
1 lubricating oil pump.
2 bilge pumps.

Crankshaft.—Of the built-up type, in two interchangeable parts.

Cylinders.—To consist of cast-iron liners of special composition fitted into a cast-iron entablature. The water jacket for the main cylinder liners to be formed by means of light castings surrounding the liners but not forming a portion of the main entablature, water joints in way of the scavenge and exhaust ports thus being avoided.

Each cylinder cover consists of a symmetrical conical-shaped loose liner of special composition, such liner forming the combustion space, an outer casing, and a small cover closing the small open end of the loose liner, and so arranged as to allow free expansion of the loose liner.

The small covers carry the fuel and air starting valves and connections for relief valves and indicators are provided.

Pistons are of cast-iron and are fitted with a separate cast-iron skirt. Piston rings are of Ramsbottom type.

The pistons are cooled by sea water supplied through telescopic tubes, which are attached to the piston and work through a specially designed stuffing box having a lantern gland and wiper ring fitted. The gear is so arranged that there are no moving water joints under pressure.

Piston rods, connecting rods, bedplate and main bearings are in accordance with usual marine practice for steam engines.

Forced lubrication is employed throughout, some appliance for filtering and purifying the oil being fitted.

Camshaft is driven from the main engine crankshaft through eccentrics, rods and suitable gearing. An air engine is fitted for moving the camshaft horizontally as required for ahead or astern running. Interlocking gear is provided to prevent the camshaft from being moved horizontally while the engine is either being started up on air or running on fuel, and also to prevent the fuel cams from being in operation while the starting cams are in action or *vice versâ*.

A steam turning gear is provided for turning the engine.

One fuel pump is provided for each cylinder.

The speed of the engine is controlled by hand gear connected to the suction valves of the fuel pump.

The lift of the suction valves can be regulated separately for equalising the power in the cylinders.

A governor is provided which operates on the suction valves of the fuel pump.

A pyrometer set is provided for indicating exhaust gas temperature at each cylinder.

Thrust block is of the Michell type.

Shafting of forged ingot steel.

Stern tube of cast-iron with a brass bush lined with lignum vitæ at the outer end.

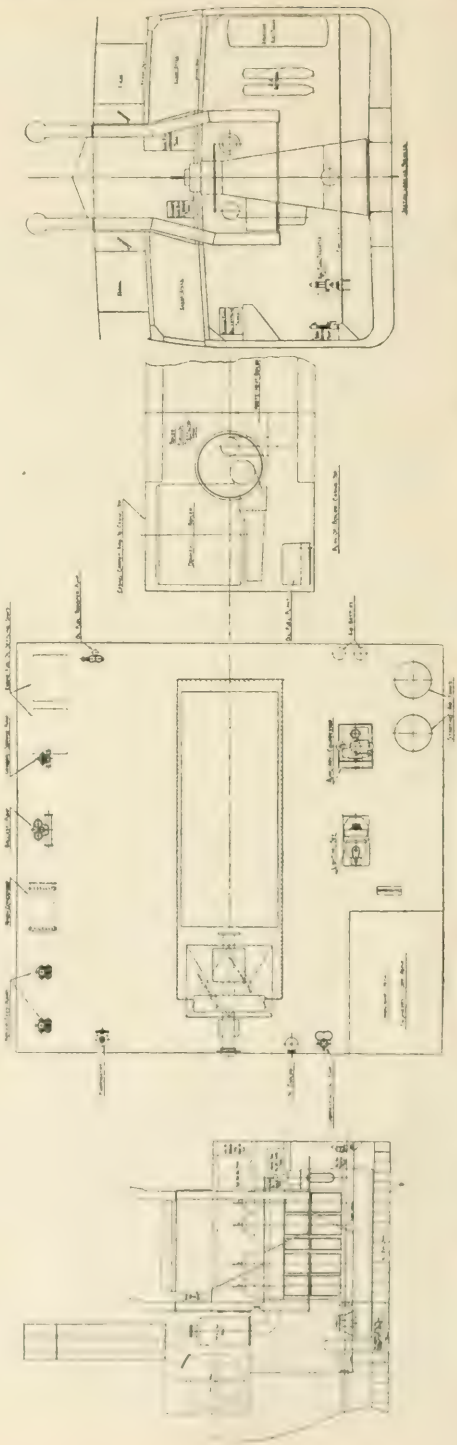
Propeller is of cast-iron.

Air bottles.—Injection air bottles capacity, 16 cubic ft. Starting air tanks capacity, 460 cubic ft.

Auxiliary machinery.—

(*a*) With steam deck machinery as specified by the Institution :—

1 ballast pump, 9 in. by 10 in. by 10 in., duplex type; acts also as standby to main engine jacket water circulating pump.



The Neptune Marine Oil Engine. Proposed Arrangement of Propelling Machinery for 371-ft. Motor-Driven Cargo Vessel.

1 general service pump, 7 in. by 7 in. by 8 in., Duplex type; acts also as standby to piston water circulating pump.
2 boiler feed pumps, 6 in. by 4½ in. by 10 in., Simplex type.
1 condenser, 500 sq. ft. surface.
1 evaporator, 10 ton capacity.
1 auxiliary compressor, capacity 140 cubic ft. of free air per minute to 1,000 lb. per square inch, driven by a compound steam engine.
1 fuel-oil transfer pump, 4 in. by 4 in. by 6 in., Simplex type.
1 standby lubricating oil pump, 5 in. by 4½ in. by 12 in.
1 donkey-boiler, 12 ft. diam by 10 ft. 6 in. long, with oil-burning installation. 100 lb. per square inch working pressure.
1 waste heat boiler (100 lb. per square inch working pressure) of special design, through which the exhaust gases from the main engine are passed, and sufficient steam will thus be raised to provide for normal requirements at sea. An auxiliary oil burner is fitted to this boiler for use when the steam supply is insufficient.

(*b*) In the event of the adoption of electrical deck machinery the following motor-driven pumps would be provided :—

1 ballast pump.
1 general service pump.
1 lubricating oil pump.
1 fuel-oil transfer pump.
1 auxiliary compressor.

also—

3 65-kw. Diesel-driven generating sets.
1 waste heat boiler for heating, &c.

The Scott-Still Marine Oil Engine.

By Scotts' Shipbuilding and Engineering Co. Ltd.

General description.—The main propelling engine consists of one set of two-stroke cycle Scott-Still marine oil engines, with four cylinders of 25 in. diam. by 45 in. stroke. Forced lubrication is supplied to the main bearings, connecting rod bearings, and guides.

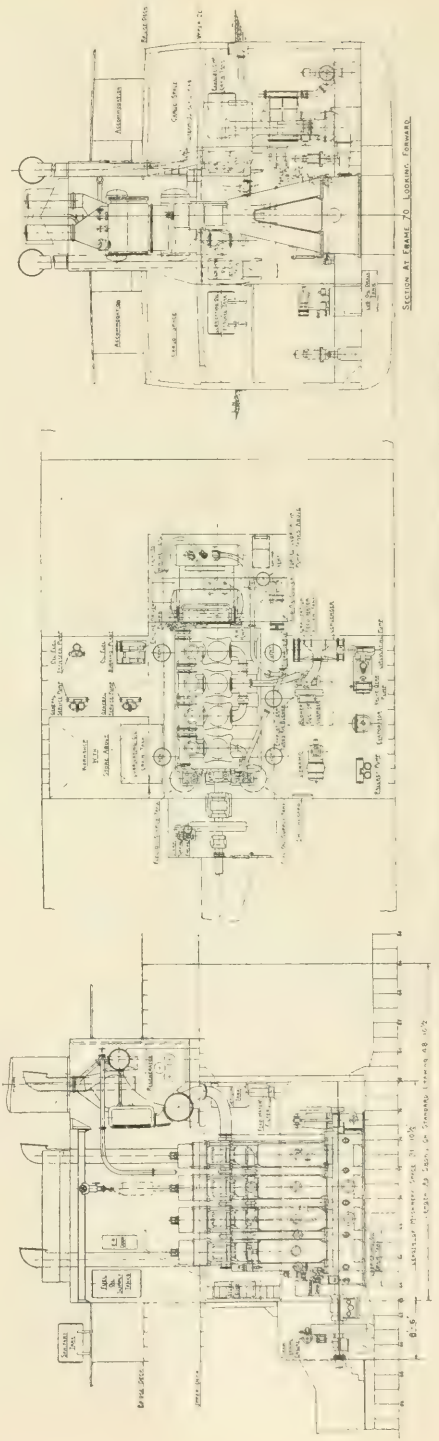
Airless injection of fuel to the cylinders is adopted, and the fuel valves are automatic in action, being arranged to open under the action of the fuel pumps. One fuel pump is provided for each cylinder, worked by a cam fixed to the engine crankshaft. Arrangements are made for running the main engine ahead and astern on oil.

Operation of the steam inlet and exhaust valves is controlled by distributors driven by gearing from the crankshaft at engine speed, and reversing is accomplished by the movement of piston valves in the distributors. A pump of Hele-Shaw type with automatic variable delivery gear, supplies oil at 400 lb. per square inch to the distributors for the operation of the steam valves. This pump is driven off the condenser circulating pump, which is arranged to suit this duty.

The scavenging air is supplied by a turbo-blower driven by exhaust steam from the main engine. The turbine is of 80 b.h.p., with a normal output of 7,000 cubic ft. per minute, and the air discharge is led to the engine bed plate for distribution to the cylinders. The air is drawn from a cowl on deck in the usual manner and a reservoir or tank, suitably drained, is interposed to arrest water or grit.

The engine has a flywheel of cast-iron with teeth on the rim to take the worm on the turning engine. The thrust block is of Michell type, arranged for forced lubrication.

A regenerator or boiler provides steam generating surface of 1,150 sq. ft., to raise steam at a pressure of 150 lb. per square inch, for starting and manœuvring the main engine, oil being supplied to the burners for these purposes. When the engine is "under way" this regenerator (the burners having been shut off) serves the purpose of a steam and water reservoir in which steam is raised, from the heat in the exhaust gases and cylinder jackets, and utilised in the main engine cylinders for the



The Scott-Still Oil Engine. Arrangement of Machinery for Single-Screw Cargo Vessel.

production of power. Three oil fuel burners are fitted and the burning plant is of Wallsend-Howden type, an extra pump being provided as a stand-by. The gases from the generator pass through a feed heater and thence up the funnel. Advantage is taken of the steam for the purpose of cooling the main engine pistons.

A group of pumps is driven by levers from the forward end of the main engine crankshaft, comprising the following:—Air pump, feed pump, forced lubrication pump, bilge pump, sanitary pump, and regenerator circulating pump. The duty of the last-named pump is to draw water from the regenerator and return it thereto through a pressure filter with the object of cleansing the water. A standby feed pump is provided of Weir's vertical type, 4 in. diam. by 6 in. by 12 in. stroke. A pump of similar type is fitted, 6 in. diam. by 13½ in. by 15 in. stroke, to take the duties, on emergency, of either the engine-driven forced lubrication pump or the valve-gear oil operating pump.

A condenser designed for a vacuum of 28 in. is provided, having 600 sq. ft. cooling surface. The condensate is effectively filtered before entering the feed-water heater. The circulating pump, already referred to, is of centrifugal type, 7 in. bore, running at 420 r.p.m., and discharges 800 gallons per minute.

The other auxiliaries included are:—Two general-service vertical duplex pumps, 7 in. diam. by 9 in. by 8 in. stroke; one ballast vertical duplex pump, 9 in. diam. by 10 in. by 10 in. stroke; one oil-fuel-transfer pump, 5 in. diam. by 4½ in. by 10 in. stroke; one electric generator giving 8 kw. at 110 volts, and a steering engine of "Wilson-Pirrie" type (not shown on the arrangement of machinery), with cylinders 8½ in. diam. by 8½ in. stroke. All the auxiliaries are steam driven. In connection with the feed "make up" a quantity of fresh water is carried on the ship's tanks amounting to 40 tons. In addition, an evaporator of 10 tons capacity per day is fitted. The winches and windlass are operated by steam supplied by the regenerator, which for this purpose fulfils the function of a donkey-boiler. For this duty oil fuel is employed in the burners.

Statement embodying the information required in connection with the Scott-Still propelling machinery, the item numbers given corresponding with those in the Institution's Specification.

1.—(*a*) The total brake horse-power of the propelling engines to enable a speed of 10¾ knots to be maintained on a four hours' trial is 1,870.

(*b*) The brake horse-power that can be maintained on a run of 3,000 nautical miles under ordinary sea conditions is 1,725.

2. An overload in brake horse-power above trial conditions of 35 per cent. can be obtained.

3. The mechanical efficiency of the engine is 88 per cent.

The bases upon which this figure is arrived at are as follows:—The mechanical friction of the engine is equivalent to 11 lb. per sq. in. of piston area (from data taken from extensive trials of a similar engine). The mean pressure equivalent to 1,870 b.h.p. at 106 r.p.m. is 79 lb. per sq. in. The engine efficiency is therefore $\frac{79}{90}$ or 88 per cent.

4.—(*b*) The consumption of fuel per brake horse-power for "all purposes" is estimated at 0·37 lb.

This is the only significant figure in the particular proposal put forward, where all the auxiliaries, including the scavenging blower, are driven by steam produced from the waste heat of the main engine.

5. The brand of fuel assumed is Anglo-American Diesel Oil, and its calorific value is 18,000 B.Th.U. net. From examinations which have taken place on various occasions after trials, admittedly of short duration (8 hours), of a similar engine, with the heavier boiler fuel oils, there is nothing to suggest that these oils cannot be used satisfactorily in the engine.

6. The total weight of the main propelling engines and auxiliaries in engine room is estimated at 375 tons. This does not include the weights of steering engine, winches or deck pipes.

7.—(*a*) The cost per ton-mile of fuel and lubricating oil is 0·00404 pence (taking fuel oil at 77s. 6d. per ton, lubricating oil at 2s. 7d. per gallon, and a speed of 10·45 knots).

(*b*) The engine-room complement is:—

Chief	1
Watch engineers	3
Stand-by engineer	1
Greaser class	6
Storeman	1
Total	12

8. The arrangement of machinery shows that the length of engine room claimed is 31 ft. 10½ in., while the length shown on the standard drawing is 48 ft. 10½ in.

The volume of the machinery space claimed is 31,800 cub. ft. less than that shown on the standard drawing.

It is further noted that the capacity of the machinery space claimed is 17,300 cub. ft. less than that based upon the stated minimum length of engine room (40 ft. 1½ in.) to give the 32 per cent. deduction. This result shows that the agitation in certain quarters for a revision of the tonnage rules is not without reason.

Vickers' Internal-Combustion Engine.

By Vickers' Ltd.

As alternative arrangements have not been invited, the proposal submitted is one in which a compromise between a low first cost and economy of running has been sought. No attempt has been made, however, to cut down the installation below the margin indicated by sea experience.

The nature of the installation and the disposition of the important items are indicated on the accompanying machinery arrangement (*see next page*). It will be observed that the total engine room length is 42 ft. 6 in., this affording ample space for all the plant required.

The main engine has been primarily designed to give continuous running at a sea speed of 10½ knots without fear of excessive wear or replacements. Attention is drawn to the fact that the mean indicated pressure in these conditions is no more than 96 lb. per sq. in., the mechanical efficiency of the engine being taken at 78 per cent., as judged by dynamometer tests of similar engines.

The four-hours' trial at 10¾ knots is easily within the capacity of the engine.

The main engine has six four-stroke cylinders, 28 in. diam. by 45 in. piston-stroke, and will develop 1,760 b.h.p. at 112 r.p.m., giving the sea speed of 10½ knots.

It is of the solid-injection type, no injection air compressors being required, thus greatly simplifying the engine and reducing the cost of upkeep. The saving of weight affords an opportunity of increasing the rigidity of the engine and allowing of ample bearing surfaces.

The aim has been to produce an engine of robust proportions throughout, easily manufactured without the use of special tools and having all parts readily accessible for examination or replacement. No attempt has been made to reduce weight or space occupied at the expense of these features.

The bedplate is flat-bottomed and directly attachable to tank top, so saving expense to shipbuilders for seatings.

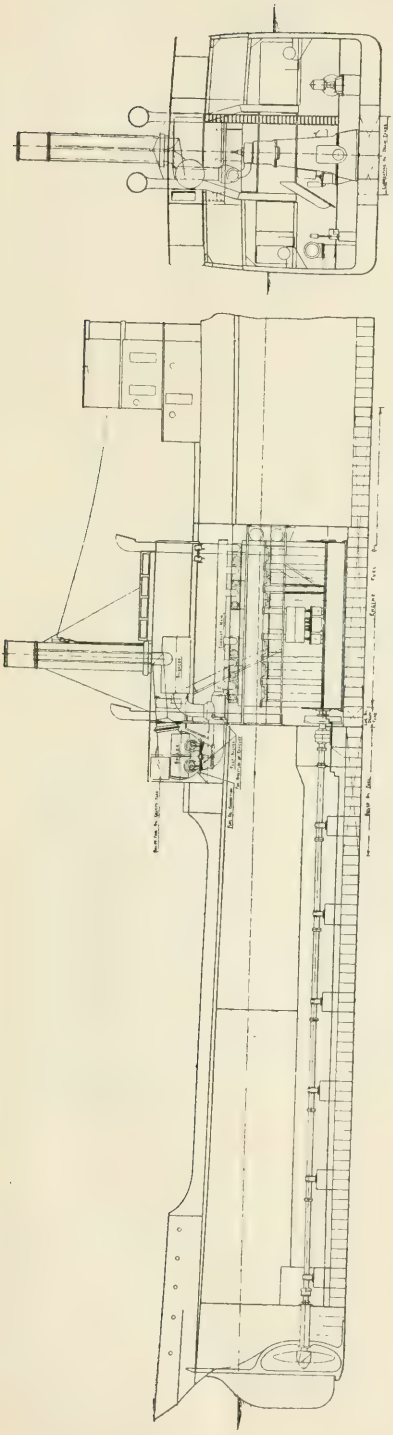
Forced lubrication is used throughout.

Salt water is used for the cooling of the jackets, covers and pistons, the circulating pumps as well as those for the lubricating oil being driven by the main engine.

These pumps are of slow-speed type to ensure durability.

A diaphragm is fitted between the crankcase and the cylinders to prevent any possible contamination of the bearing oil, thus economising lubricating oil.

The exhaust valves in the cylinder covers are water-cooled. This type of valve has proved very successful in the "Vickers" engine, and it has been recorded that during 18 months' running of a twin-screw installation comprising 16 cylinders,



The Vickers Engine. Arrangement of Machinery for Single-Screw Vessel, 1890 b.h.p.

only one exhaust valve was opened up, and this only to ease the spindle.

Judging from trials carried out in Vickers-built motor vessels, the exhaust gases from the engine would, if passed through the boiler, generate sufficient steam for all auxiliary purposes at sea; but as there is insufficient margin to cover slow-speed running conditions, the dynamo is driven by a semi-Diesel engine.

The boiler is designed to cover the maximum requirements of steam-driven deck auxiliaries and for either exhaust-gas or oil firing, the furnace fronts being arranged for rapid conversion.

For extensive manœuvring a steam-driven compressor is fitted, a semi-Diesel-driven auxiliary being provided for emergency or sea use. Generous compressed-air storage capacity is provided.

The oil engines for the dynamo and auxiliary air compressor are arranged in line so that there is a spare engine for the dynamo available at short notice.

The propeller is right-handed and of the built-up type. The boss is of cast-steel and the blades of manganese bronze. It is considered that the superior efficiency of this type of propeller would soon repay the extra cost over one of cast-iron.

The thrust block is of the Michell patent single-collar type.

The shafting generally, in common with all other parts of the installation, is proportioned to conform to the requirements of Lloyd's Register, and the weights given are calculated on that basis.

The fuel consumption for the main engine is estimated at 0·42 lb. per brake horse-power per hour, and for all purposes at 0·43 lb. per brake horse-power per hour. The calorific value assumed is in the neighbourhood of 18,000 B.Th.U.'s. The engine could be adjusted, if desired, to function on the more difficult fuels, but such are not recommended, as entailing greater mechanical wear on any engine and greater upkeep. Fuels of Texas quality, light Mexican, and similar fuels not containing large quantities of impurity—especially hard asphalt—are eminently suitable when Diesel fuel is not available.

The weights of the machinery are given below:—

The cost of running per ton-mile would be as follows, on a deadweight tonnage basis of 6,540 tons:—

Items included:—Chief engineer and three assistants, 3 greasers, 1 donkeyman, 1 storekeeper. Wages and Food. Fuel cost at £4 12s. 6d. per ton. Lubricating oil cost at 3s. per gallon. (Total running cost per ton-mile 0·0067 penny.)

Approximate Machinery Weights.	Tons.
Main engine complete with engine-driven pumps, flywheel, electrically-driven turning gear, all piping and connections attached, also holding-down bolts and chocks	270·0
Air reservoirs, exhaust silencer, lubricating-oil settling tank, cooler, and strainers, fuel oil ready-use tank mountings and strainers	20·5
Air compressors and oil-driven dynamo	20·5
Shafting complete with all accessories	48·0
Ladders and gratings, floor plates and bearers, lifting gear, sundry bolts, nuts, joints, clips, hangers, engine-room ventilators, telegraph gear, spanners, special tools and outfit	28·0
Piping and connections clear of engine, including strainers and sundry fittings	27·5
One single-ended cylindrical boiler 10 ft. diameter by 11 ft. long, with steam-driven auxiliaries for ballast, general service, fuel-oil transfer, &c., and with uptakes, funnel and steam and exhaust pipes in machinery space	32·5
Water in engine, boiler and pipes affected	20·0
Total	460·0

The Werkspoor Engine.

By R. & W. Hawthorn, Leslie & Co. Ltd.

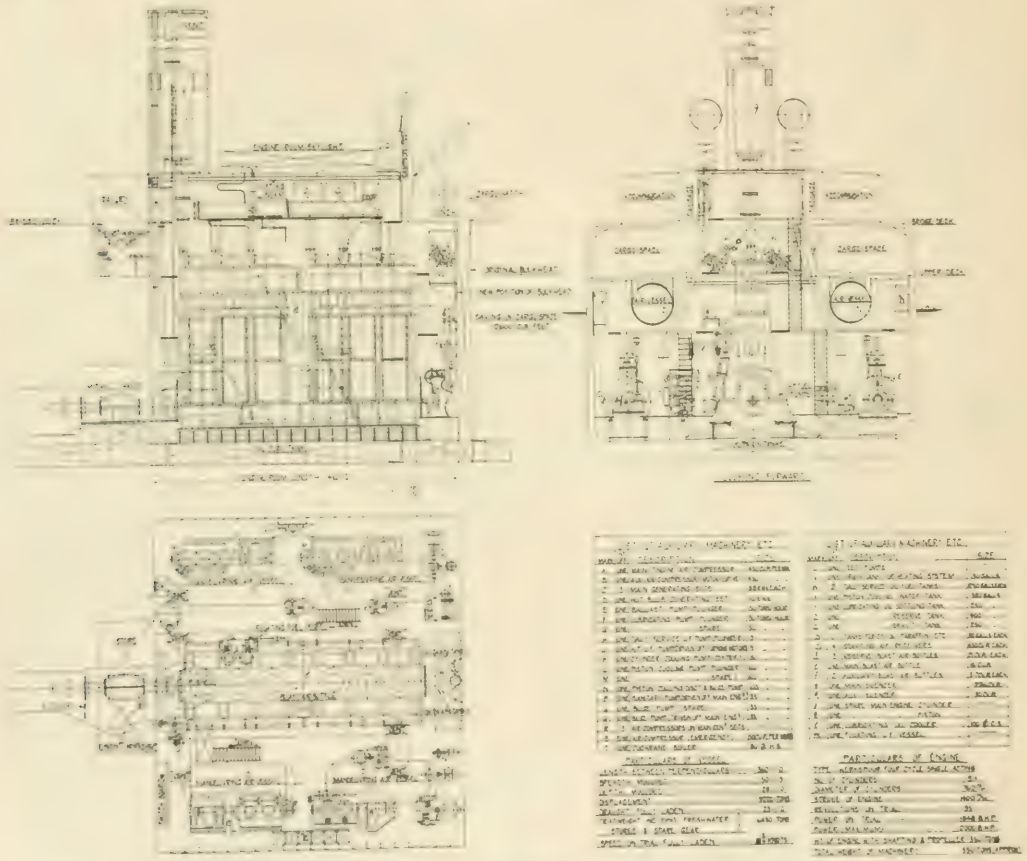
The following particulars are given regarding a machinery installation embodying the Werkspoor heavy marine oil engine suitable for propelling the review ship under the conditions laid down by the Institution.

The engine selected in conjunction with the review ship is moderately rated, and in fact is able to maintain trial-trip conditions continuously at sea.

The engine is of the six-cylinder four-stroke single-acting type, having a cylinder diameter of 760 mm. and a stroke of 1,400 mm., together with a speed of revolutions of 95 at maximum output of 2,000 b.h.p. on brake. At a speed of revolutions of 95 per minute, and using a propeller in accordance with the "revolutions power curve" issued with the plans of the ship, the brake horse-power necessary is 1,848, when propelling the ship at 10¾ knots under trial-trip conditions of draft. The engine would develop this power with a mean indicated pressure of 90 lb. per sq. in., and would, therefore, be running under very safe conditions. The auxiliary machinery is electrically driven, power being provided by three Diesel generating sets of 88 kw. each and one semi-Diesel generating set of 10 kw. One electrically operated auxiliary compressor is fitted, equal in capacity to the air compressor driven off the main engine. At sea one Diesel generating set will be sufficient, whilst, when manœuvring, two sets will be required for providing starting air. In port the maximum load will require two sets in operation, thus leaving at all times one stand-by set available in case of emergency.

A small Cochran boiler of about 80 sq. ft. of heating surface is fitted for accommodation heating and also for the heating of oil fuel in the double bottoms and in the daily service tanks when required.

The following answers are given to the questions selected by the Institution :—



The Werkspoor Engine. Machinery Arrangement with Electric Auxiliaries for Single-Screw Cargo Vessel.

1. The engine will develop 1,848 b.h.p. when the ship is travelling at 10¾ knots during a four-hours' trial.

2. The percentage of the trial brake horse-power which can be maintained by the proposed engine on a 3,000-mile run under ordinary sea conditions is 100 per cent., inasmuch as the engine is capable of developing 2,000 b.h.p. if required.

3. About 73 per cent. is the mechanical efficiency of the engine, which drives its own air compressor, bilge and sanitary pumps, together with the high-pressure fuel pumps.

4. The fuel consumption of the main engine per brake horse-power when the ship is travelling at the designed speed is 0·41 lb. per brake horse-power per hour, or 8·1 tons of oil fuel per 24 hours. The total consumption for all purposes,

under the same conditions, is 0·45 lb. per brake horse-power per hour, or 8·9 tons of oil fuel per 24 hours.

5. The brand of fuel assumed is that regularly supplied for marine oil engines, and has the following specification, viz.: Specific gravity, 0·86 to 0·93 at 60° Fahr. calorific value 17,800 B.Th.U. per lb. low value. Flash point 160° Fahr. (Abel Pensky close test). Viscosity 155 second Redwood at 120° Fahr. Sulphur content and ash content 1 per cent. maximum by weight respectively. Oils conforming to the above specification are suitable in every way for continuous running.

The grades of boiler oils which have been used recently for Diesel engines can be burnt successfully, and other less refined boiler oils can be also used, but due to their ash and hard asphaltum content, together with earthy matter, are not to be recommended on account of the abrasive and clogging action which takes place within the cylinder, together with pitting of the exhaust valves, all of which make for more frequent and more expensive repairs, greatly reducing any economy that might have been anticipated by using the cheaper grades of oil fuel.

6. The weight of the main engine, including intermediate and propeller shafting, and propeller, is about 350 tons. The total weight of machinery, which includes generating sets, auxiliary pumps, pipes, valves, platforms, gratings, silencers, spares, &c., is, approximately, 550 tons.

7. The length of engine room required is 46 ft. 9 in. The saving in cargo space is therefore 2,600 cubic ft. over that given in the plan of the review ship. A small thrust recess has been added to the original engine room.

8. The cost of fuel and lubricating oil per ton-mile is based on an average price of 80s. per ton, and 3s. per gallon respectively. With a speed of 10¾ knots at full load, the distance travelled per day is 258 nautical miles, the estimated fuel consumption being 8·9 tons per day and the estimated lubricating-oil consumption 12 gallons per day. For comparative purposes, the deadweight carried is 6,450 tons, corrected for machinery weight; therefore the fuel and lubricating-oil cost per ton-mile is 0·00539 pence.

The number of the engine-room staff is eleven, including four engineers, one electrician, and six assistants.

SOME USEFUL NOTES ON THE CLEANING OF MARINE TURBINES.

It is often found, from the record of pressures taken, that silting up or choking is taking place in the blading of marine turbines; consequently, the efficiency of the machinery falls off rapidly as the amount of choking increases. When this condition is noted, it may not be convenient or even possible to open up the turbines for hand cleaning.

The principal causes of choking in the turbines are, first, sediment carried over with the steam from the boilers, and secondly, grease or oil becoming introduced. Very often the effect is due to a combination of the two causes.

Dealing with the first, it is to be remembered that with turbine machinery vacuum is important and at all times should be maintained at the highest for which the turbine is designed. The demand on the boilers for steam is constant, and, owing to the vacuum, insistent; consequently, there is considerable inducement for priming to take place, and water and sediment to be carried over with the steam. This sediment, or at least some considerable portion of it, inevitably adheres to the turbine blading.

Many forms of mechanical separators or steam purifiers have been introduced to cope with this trouble, some of them have proved very effective in service, and have reduced, almost to vanishing point, the amount of water and sediment carried over from the boilers.

The composition and quantity of the sediment in the boilers will, however, vary with the water used and the treatment adopted, but in all cases there is, in the sediment formed, a considerable amount of light, finely divided or flocculent matter which floats on the surface of the water and is picked up with the steam and carried over in spite of the mechanical means used to intercept it.

The second cause referred to is the introduction of oil found on the turbine blading in the form of a greasy deposit. Oil finds its way into the turbines from two sources usually, the first being by the introduction of surplus auxiliary exhaust steam to the low-pressure turbine; the second source is by means of lubricating oil being drawn along the shaft and in through the turbine glands. This latter source is a very common one and is due to lack of attention to the sealing of the glands; this most often occurs when fires are being cleaned, and in consequence the steam pressure has fallen more or less. At this time the attention of the engineer on watch is principally devoted to getting steam up; and the supply of steam to the turbine glands, which is usually passed through a reducing valve and so regulated as to maintain a pressure of one to two pounds at the glands, may cease altogether, with the result that, at any rate on the low-pressure turbine, the vacuum draws the oil along the shaft. The quantity passed, of course, depends upon the condition and arrangement of the baffles fitted. The grease adhering to the blading naturally collects any deposit carried over with the steam, and this rapidly grows in the course of time.

Where the circumstances are such that (after verifying the correctness of the pressure gauges and being satisfied that the blading is choked) it is not possible to open up the turbines for cleaning, satisfactory results can be obtained by the application or introduction of turpentine and Atlas preservative.

On the day of arrival in port, after the ship is moored and before the machinery is finally shut down, shut off or blank the discharge of the air pump to the feed-filter tank and arrange for the pump to discharge to the bilges. Or, if this time is not a suitable one for the operation, the turbines should be heated up, say, on the day before sailing, and the air pump, as already stated, arranged to discharge to the bilge. Run the turbines under steam slowly, say 10 to 15 r.p.m. of the propeller shaft, having a vacuum of 15 to 20 in. at the condenser.

Heat up the turpentine to about 100°, connect a small pipe, such as is used for pressure-gauge connections, to a gauge cock on the high-pressure turbine as near the inlet end as the effect of the vacuum is obtainable. Allow half the quantity of turpentine, mentioned hereunder, to be *slowly* drawn into the revolving turbine. After this quantity has passed into the high-pressure turbine, pass the remainder of the turpentine in a similar manner into the ahead inlet-end of the low-pressure turbine, or after end of the intermediate-pressure turbine, where three turbines are fitted in series.

After the total quantity of turpentine has been introduced as described, have heated up two-thirds of the quantity of Atlas preservative and pass this into the early stage of the high-pressure turbine at full strength. The remaining quantity of preservative should be diluted with five times its quantity of boiling water, and be allowed to follow through.

It must be emphasised that the liquids should be drawn into the turbines very slowly.

Samples of the air pump discharge should be collected from time to time for inspection.

Quantities recommended for ordinary cases:—

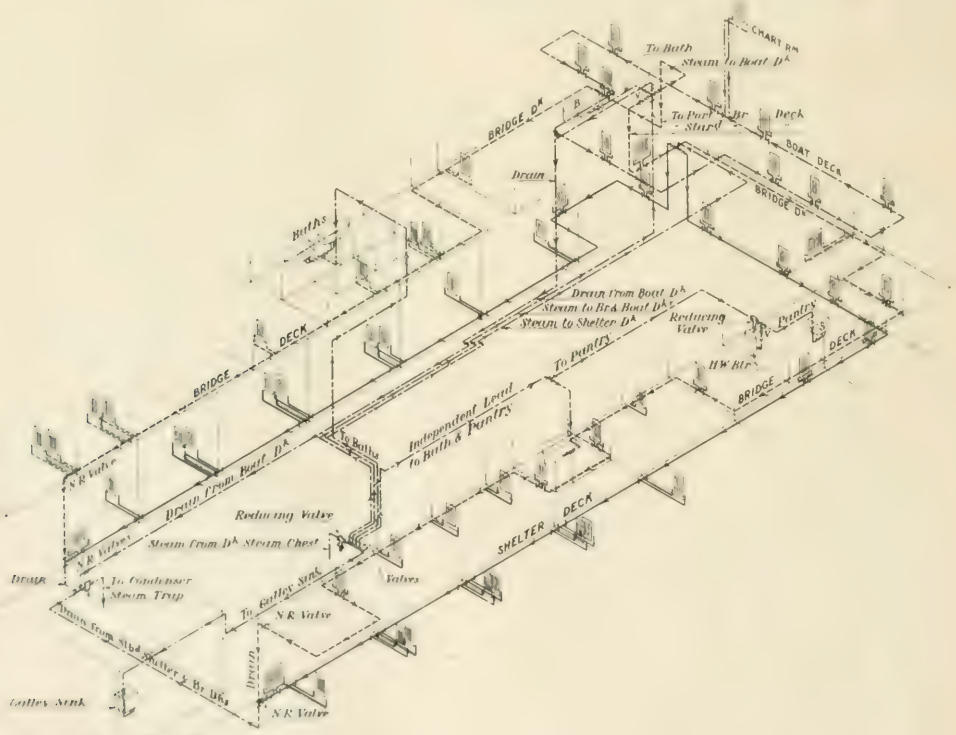
Single-Reduction Geared Turbines.—Turpentine: 2 gallons per 1,000 s.h.p.; Atlas preservative: 6 gallons per 1,000 s.h.p.

Double-Reduction Geared Turbines.—Turpentine: 1½ gallons per 1,000 s.h.p.; Atlas preservative: 4½ gallons per 1,000 s.h.p.

A SINGLE LINE HEATING SYSTEM.

The Working Principle and its Application.

The successful distribution of heat to cabins on shipboard is attended with difficulties. One of the complications in the ordinary systems of steam heating employed on many ships is the great length of piping required for supply and return purposes. Inconvenience is sometimes experienced, too, by the occupants of adjacent cabins, owing to the difficulty of cutting off the steam supply to any particular radiator.

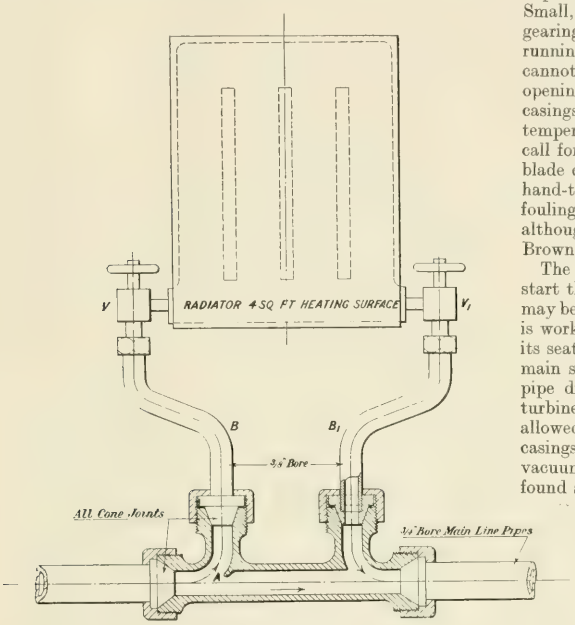


Isometric Projection, showing Outlay of Ship's Heating Arrangement on the Single Line System.

A rather ingenious device has been recently placed on the market by George Clark & Sons (Hull) Limited, which dispenses with the necessity for return piping, and allows of individual radiators being cut off as required. We show a sectional detail elevation of this on page 109. The method of working is a simple one, as will be seen. Steam flows in along the pipe from the source of supply, and enters the casting which is in the form of a converging nozzle. This has the effect of a venturimeter upon the steam particles, causing an increase of pressure and the consequent flow of steam up the left-hand curved pipe B, whence, by means of the valve V, it enters and flows through the radiator. Leaving the radiator by means of the valve V and the pipe B, it returns again to the main pipe line. If, of course, the valves V and V_1 are closed, the radiator is cut off and steam is obliged to flow directly along the pipe. By this means all return piping is dispensed with, and the system has proved economical in its working, due partly to the smaller amount of pipe surface. In the other systems, in some cases the amount of surface given by piping inside the cabins is more than the actual surface of the radiator, but in this system the amount of surface is about half, thus giving better heating control in the cabins.

The above figure is an isometric projection of a typical heating system fitted on shipboard. It will be observed, on examination of the drawing, that the steam supply is taken forward and branches off to the various groups of cabins. After supplying and draining the radiators, the pipes are reconnected and drained through a trap to the auxiliary condenser or cascade. Each section of heating is controlled by a valve which enables the whole system to be well sectionalised, and non-return valves are fitted where the drains are collected, to prevent any steam getting into a section already isolated. The steam supplies to the pantry, galley sinks and baths, &c., are on independent lines, thus enabling a full supply of steam to be secured without any steam passing through the accommodation heating system.

From the foregoing it will be apparent that the cost of installing this system is considerably lower than the cost of the double-line system since fewer holes require to be cut in the ship's structure, and the pipes can be more neatly stowed in the cabins, while the number of joints is considerably reduced. The supply pipes are of the same bore throughout, and the small pipes to the radiators are of uniform size, the supply pieces to each radiator being standard throughout the system, a feature which saves a great deal of labour in detailing, making, collecting and assembling.



Detail Diagram (not to scale) showing Method of Operation.

CONVERTED WARSHIPS.

A Correspondent writes :—The announcement in your columns for January that the German ex-coast defence ships, converted into cargo vessels, have proved very satisfactory is interesting inasmuch as real successes from the conversion of warships to merchant vessels seem to be the exception rather than the rule. It would be, therefore, of considerable interest to those concerned with shipping could some further details of the performance of these vessels be made public. They are propelled presumably by internal-combustion engines, and if this is the case, then their performance is certainly another "feather in the cap" of this type of machinery. The conversion of obsolescent and obsolete warship tonnage is a matter which has exercised the minds of practical naval architects to some considerable extent during the past few years. Thus we have seen destroyers converted into passenger ships, an old armoured cruiser running between New York and Bermuda, and a small monitor used for oil carrying purposes. Possibly the most convertible vessels were the "Kil" patrol boats, many of whom spent the greater portion of last year laid up in the Tyne, and the small shallow draught tunnel mine sweepers.

SOME NOTES ON THE PREPARATION OF TURBINE MACHINERY FOR SEA.

The time required to warm through initially a set of turbines depends largely on the size of the installation and the type. Small, high-speed sets, transmitting through double-reduction gearing, require three to four hours' preliminary heating before running, while large direct-driven and single-reduction turbines cannot safely be steamed under eight to twelve hours from opening the heating connections. Heating up of rotors and casings should be gradual and uniform so as to minimise temperature stressing and condensation. Parsons' turbines call for especial care in this direction on account of the small blade clearances. By periodically moving the rotors with the hand-turning gear when "warming through," all risks of fouling, due to uneven expansion of the blading, are avoided, although this precaution is not necessary when dealing with Brown-Curtis, Rateau, and other installations.

The first operation when warming through turbines is to start the main circulating pump so as to expel any air which may be present from the condenser. While the main circulator is working slowly, the main boiler stop valve can be eased off its seating and a small quantity of steam allowed to enter the main steam range, care being taken to ensure that all steam pipe drains are open. The ahead and astern high-pressure turbine regulator valves can now be opened and the steam allowed to filter through both high-pressure and low-pressure casings. The main air pump can now be started up and a slight vacuum maintained. A vacuum of about 4 or 5 in. will be found sufficient to keep the condenser just clear of water and water vapour, at the same time promoting the thorough circulation of the heating steam. It will be appreciated that were a much higher vacuum maintained the steam would rapidly pass straight through the turbines in parallel lines without doing its work efficiently. The steam to glands should not be opened at this stage as this would cause the vacuum to rise, with the result referred to above.

When steam has been circulating for some time, all nozzle valves, ahead and astern, should be opened and the air pump speeded up gradually. A very ready way of ascertaining whether the turbines require further preliminary heating is to watch the sliding feet at the forward end of each casing. Obviously, when the "reference mark" on the turbine foot is opposite the corresponding mark on the guide expansion of the cylinder warming-up is complete and the turbine is ready for steaming. When this stage of the operation is almost reached the forced lubrication pump can be started and the steam to glands opened a little. While the vacuum is rising the main circulator should be speeded up and all lubrication system filters, cocks, and indicators examined. All air vents on turbine and gear-case bearings (when fitted) should be opened and the oil allowed to commence circulating. When all air has been vented out of the bearings the oil supply should be so adjusted as to give a pressure of about 5 lb. per square inch at the bearings. When the thrust has been examined, and when it has been ascertained that the screw is quite clear, the machinery is ready for service, after a few preliminary revolutions of the propeller.

Trials of Marine Oil Engines.—A Committee consisting of representatives of the Institutions of Mechanical and Marine Engineers and Naval Architects, has been appointed with the objects of carrying out tests on oil engines and of reporting on the performance of motor-driven vessels. The Engineer-in-Chief of the Fleet has, with the approval of the Admiralty, joined this Committee. In scope, the proposed trials will include economy and thermodynamic tests ashore, and manœuvring tests at sea. Wherever possible the behaviour of the propellers and the hull will be examined. It is intended to test engines of as many representative types as possible.

THE USE OF ECONOMISERS IN THE MERCANTILE MARINE.

By T. A. CROWE, M.Sc.

Early Difficulties—The Case for the Economiser in the Modern Ship—Gain in Efficiency—Weight.

The first economiser known to have been used at sea was fitted by Mr. Ebenezer Kemp in the steamer *Plantyn* in the year 1879. It consisted of a plain pipe, 5 in. diam., fitted in the uptake and having a heating surface equal to about 3½ per cent. of the main boiler heating surface. With this simple apparatus he obtained a rise in temperature of 15° Fahr. and a reduction in coal consumption of 4 per cent. Larger heaters were fitted in succeeding ships, until, with the steamer *Taormina*, the economiser surface amounted to 10 per cent. of the boiler heating surface. Mr. Kemp then decided that instead of the economiser heating surface being a small fraction of the boiler heating surface, it should be greater than it. The last ship fitted with economisers by him was the *Caloric*, built in 1887. In this ship the heating surface of the boiler was 1,613 sq. ft. and that of the economiser was 3,505 sq. ft., *i.e.*, rather more than double. The inlet temperature of the gases was 675° Fahr. and the outlet temperature was 225° Fahr., which raised the temperature of the feed water from 130° to 270° Fahr. This corresponds to a rate of heat transmission of only about 300 B.Th.U. per square foot of heating surface per hour. The coal consumption of this ship was very carefully measured on several occasions under ordinary sea conditions in the Atlantic and Mediterranean trades. Each trial lasted 24 hours and the coal was accurately weighed. A coal consumption of 1¼ lb. per indicated horse-power was obtained, which must be considered a remarkable performance for a ship of this period using saturated steam at a pressure of 160 lb. per square inch. In comparison with an exactly similar ship without economisers the *Caloric* showed a saving in coal of fully 15 per cent.

Early Difficulties and their Solution.

The economiser fitted in the *Caloric* possessed two great faults which prevented its general adoption. Firstly, it was too bulky and heavy, and secondly, it suffered very much from corrosion and deposition of sediment. These two faults were intimately associated with each other. The great volume of the economiser caused the flow of the feed to be so sluggish that air bubbles were formed on the surface which caused pitting; and if there was any sediment in the feed water, the speed was not sufficient to maintain it in suspension, and this, in turn, caused local overheating. Mr. Kemp also had trouble with corrosion caused by using salt water for make-up feed, the use of tallow on the tail rods, and too much oil for cylinder lubrication.

In spite of this long list of troubles none of them will be found to be insuperable. Considering each in turn: the first and greatest difficulty of bulk and weight can be overcome by the use of high gas speeds and the substitution of steel for cast-iron. Messrs. Royds and Campbell found in one of their experiments* that by using a high gas velocity (corresponding to a draught of 4 in. of water) and a gas inlet temperature of 500° Fahr. and an outlet of 260° Fahr., and a water inlet of 65° Fahr. and an outlet of 149° Fahr., that they obtained a heat transmission rate of 5,040 B.Th.U. per square foot of heating surface per hour, *i.e.*, more than 16 times the rate of heat transmission obtained by Kemp with his bulky economiser and low gas speeds. Substituting constants derived from their experiments in Reynolds' Law of Heat Transmission,"

* Trans. of the Inst. of Engineers and Shipbuilders in Scotland, Vol. LV.

Messrs. Royds and Campbell found that with a ratio of

$$\frac{\text{Economiser heating surface}}{\text{Boiler heating surface}} = 0{\cdot}4,$$

a gas inlet temperature of 500° Fahr., and an outlet temperature of 268° Fahr., and a water inlet of 180° Fahr. and an outlet of 280° Fahr., they could obtain a heat transmission of 2,020 B.Th.U. per square feet per hour. This economiser would require a draught of about 4 in. of water, but this would not result in a higher rate or combustion in the boiler furnace, as the extra 4 in. of draught would only be required to overcome the resistance of the economiser to the flow of gases.

The trouble due to corrosion and sedimentation in the early economisers can be largely prevented by making the economisers smaller and using more rapid circulation. This would prevent the sediment from being deposited, and would not allow air bubbles to collect. But, if possible, all feed water should be de-aerated before reaching the economiser, either by means of a preliminary exhaust-steam feed heater, or by the use of a closed feed system. In addition to the above, the quantity of oil used for internal lubrication in the case of superheated steam should be reduced to a minimum and eliminated in the case of saturated steam. Feed-water filters should be maintained in an efficient condition, and only distilled water used for make-up feed. If all these points are attended to there should be no trouble from excessive corrosion.

The Case for an Economiser in a Modern Ship.

The advantage of fitting economisers at the present time is much greater than it was in the time of Kemp. About the year 1880 most ships were fitted with compound engines which used steam at a pressure of about 60 lbs. per square inch (corresponding to a temperature of about 308° Fahr.). This comparatively low temperature enabled the temperature of the hot gases leaving the boiler to be reduced to about 450° Fahr. and still get a reasonable rapid heat transference per square foot of heating surface. It must also be remembered that the majority of ships at this time depended on natural draught in which case it would not be advisable to cool the funnel gases below 450° Fahr., which would only give a draught of about ⅜ in. water pressure. Therefore, practically no heat was available for use in the economisers, and if one was fitted, it obtained its heat either at the expense of the boiler or the draught. The case for economisers in a modern boiler room is very different. Steam pressures of 200 lb. to 225 lb. per square inch are in general use, which means a steam temperature of nearly 400° Fahr., and therefore, in order to get the same temperature gradient in the boiler plates, the hot gases must leave the boiler at 550° Fahr., *i.e.*, 100° Fahr. higher. Forced draught is now the general practice except in the case of the smallest tramp steamers, therefore there would be no difficulty in obtaining the greater draught pressure necessary to obtain a high gas speed through the economiser. The additional power required from the fan engine would only amount to about 3 per cent. of the heat in the fuel.

Gain in Efficiency due to Economisers.

Suppose a ship is fitted with a feed-water heater supplied with exhaust steam from the auxiliary machinery, which

raises the feed temperature to 160° Fahr. and in this way eliminates practically all the air from the feed water.

Assume :—

1. Temperature of gas entering economiser .... 550° Fahr.
2. Temperature of gas leaving ecomomiser .... 260° Fahr.
3. Lbs. of gases per lb. of coal burnt .... 22 lb.
4. Specific heat of funnel gases 0·24
5. Temperature of water entering economiser .... 160° Fahr.
6. Lbs. water evaporated per lb. of coal burnt .... 8·5 lb.

Let t = temperature of water leaving economiser, then

$$8 \cdot 5\ (t - 160) = 22 \times \cdot 24\ (550 - 260)$$

$$\therefore t = 342° \text{ Fahr.}$$

That is, the feed water will enter the boiler at a temperature of 342° Fahr., or only 46 deg. below the steam temperature, thus allowing the boiler to perform its proper function of a steam generator and not that of a feed-water heater. This evaporation corresponds to a boiler efficiency of 65 per cent. if the calorific value of the coal is 14,000 B.Th.U. If the feed temperature is raised to 342° Fahr. there will be a gain in efficiency of nearly 17 per cent. due to the economiser, and the overall boiler efficiency will be raised to 76 per cent.

If the boiler is of the cylindrical type, working at 200 lb. per square inch, and is fitted with a smoke tube type of superheater supplying steam at 200° Fahr. superheat, the steam temperature will be 588° Fahr. Reducing the temperature drop in this case to 100 deg. between the hot gases and the superheated steam, this gives a temperature of 688° Fahr. for the hot gases leaving the boiler and entering the economiser. If no economiser is fitted this results in a reduction of the boiler efficiency by 5 per cent. to 60 per cent., and the evaporation per lb. of coal will be reduced to 7·2 lb. The temperature of the gases leaving the economiser in this case will be fixed by the maximum feed water temperature entering the boiler, which should not exceed 380° Fahr. With the above exceptions the assumptions will be the same as before.

Let t = temperature of gases leaving the economiser, then

$$7 \cdot 2\ (380 - 160) = 22 \times \cdot 24\ (668 - t)$$

$$\therefore t = 386° \text{ Fahr.}$$

A gas outlet temperature of 386 deg. would leave sufficient heat for raising the forced draught temperature to 200° Fahr. The gain in efficiency in this case would be 19 per cent. due to the economiser and 7½ per cent. due to the air heater, and would result in an overall boiler efficiency of 76·0 per cent.

It is found in practice that there is a small additional gain in efficiency compared with what might be expected due to the easier steaming of the boilers with hot-feed water.

The following table has been calculated to show the gain in efficiency which might be expected through fitting economisers with varying temperatures of the hot gases leaving the boiler.

Economiser Temperatures, ° Fahr.				Lbs. of water per lb. of coal	Boiler Efficiency.		Per cent. gain in Efficiency due to Economiser.
Gas.		Water.			Without Economiser.	With Economiser.	
Enter.	Leave.	Enter.	Leave.				
450	260	160	270	9·1	63	76	10·2
500	260	160	306	8·8	65	76	13·5
550	260	160	342	8·5	67	76	16·7
600	260	160	378	8·2	69	76	20·0
688	386	160	380	7·2	60	71·5	19*

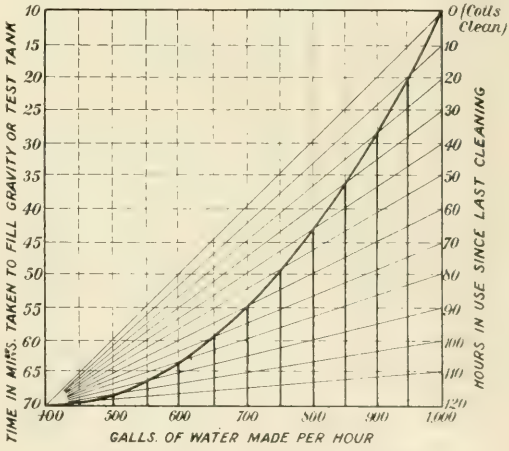
*Including air heater.

Weight.

Finally, there is the vexed question of weight. Consider the case of an ordinary tramp steamer of about 10,000 tons d.w. and 3,000 i.h.p. She will probably be fitted with three boilers of about 2,500 sq. ft. heating surface each. Basing the economiser design on the use of high gas speeds, and making its heating surface 0·4 of the boiler heating surface, then the weight of each economiser would be about 5 tons, *i.e.*, about 10 per cent. of the weight of the boilers, empty and without fittings. Suppose the above ship burnt 48 tons of coal per day without economisers, then, when fitted with them, she would only burn 39·8 tons, or on a voyage of 20 days there would be a saving of 164 tons. Surely it is worth while carrying 15 tons weight of economisers in a ship to save 164 tons of coal on each voyage. If the ship was at sea for 200 days each year and the average price paid for bunker coal was 30s., there would be a saving of nearly £2,500 per annum in the coal bill, which would be more than sufficient to cover extra depreciation and interest on capital.

A USEFUL GRAPH FOR DETERMINING TIME TAKEN TO FILL GRAVITY TANKS.

The data given in the diagram on this page are intended purely as a guide in assisting those in charge of evaporators to determine the time and output under different conditions of coil cleanliness. The actual figures are supplied from previous trials, and depend, of course, on the type and capacity of the distilling plant. The horizontal reading gives the number of gallons of water made per hour under the varying conditions. The right abscissæ give the number



Graph showing Time taken to Fill Tanks.

of hours the evaporator has been in use since the last cleaning of coils, and the left gives the minutes taken to fill gravity or test tank. This is a great help, since the gravity and test tanks are usually situated in the engine-room casings right away from the starting platform. The resultant curve plainly shows the drop in efficiency due to dirty coils.

We are asked to draw the attention of our readers to the fact that in the advertisement of the White Patent Oil Burning Company, published in the February issue of *The Marine Engineer*, the reference to the "1 I.H.P. obtained per lb. of oil in daily service," should read "1 I.H.P. obtained per 0·73 lb. of oil in daily service."

THE CARE AND OPERATION OF WATER-TUBE BOILERS—II.

Tubes—Feed Water and Feeding—Addition of Lime—Care of Boilers after Use—Examination.

It frequently happens that some furnaces have a much better draught than others; the wing furnaces usually suffer in this respect. In order to keep fires at a uniform thickness the draught plates should be regulated, and fires which have more draught will require more coal to keep the whole grate area level. The forced draught fans should also be adjusted to ensure steady steam output. Where boilers are fitted with special appliances for extra air supply, use should always be made of such fittings. The ash pans must never run short of water, and there must at all times be sufficient water in them to prevent the fire bars, and also the pans themselves, from being burnt. It is very undesirable with water-tube boilers to allow the steam to escape through the safety valves, and, should it be necessary suddenly to check the output of steam, the draught plates must be closed, fans eased down, and the furnace doors very gradually opened. In the case of Belleville boilers, open the smokebox doors. It is considered inadvisable to use an excessive amount of feed water to lower the pressure, as this is almost certain to cause priming when the engines are started again. The silent blow-off valve should at all times be very carefully used. As regards cleaning the fires, that portion opposite (one fire door on each boiler) should be cleaned during one watch, that portion opposite the next fire door during the next watch, and so on. It is unnecessary and uneconomical to keep banked fires in water-tube boilers, especially if the water in the boilers is still hot, but if such boilers are likely to be required quickly the fires should be laid.

Tubes.

The fire side of the tubes in a water-tube boiler require careful cleaning when under steam, this operation being quite as important as sweeping tubes in a cylindrical boiler. As a general rule it is a good plan to clean the tubes before cleaning the fires, in order to prevent the soot and deposits falling on the clean fire and dulling it. The frequency of cleaning will depend on the quality of the coal being burnt and also on the power required from the boiler. Tubes are usually cleaned by means of a steam or air lance, or in some water-tube boilers by a fixed apparatus. The lance must be thoroughly blown through before operating in order to prevent moisture mixing with the deposit and caking on the tubes. The lance is worked in a horizontal direction, starting at the top and working from side to side down to the bottom. Air nozzles are sometimes fitted to clear accumulations from the slopes of the uptakes, and should always be used for the purpose intended. After cleaning, the speed of the fans should be increased for a few minutes to blow the loose dirt up the funnel. Yarrow boilers are provided with portable doors in the front casing, through which the lance is inserted and worked over the whole surface of the tubes. The deposits which form on the lower rows of the tubes should be removed by means of scrapers during the operation of cleaning fires. After cleaning a boiler internally and externally a coating of blacklead should be applied to the interior of the steam and water reservoirs and to the exterior of the tubes, the latter preventing the deposition of scoriæ on the outside of the tubes when steaming.

Feed Water and Feeding.

When under way the water level should never be above half a glass, and it will be found that one-third gives the best result. Automatic feed regulators are usually fitted with an arrangement by means of which the working level can be adjusted. In some cases an adjusting screw is fitted, but in the Belleville a number of weights are used. At least once a watch the automatic feed valve should be moved by hand to ensure that the float is working freely and accurately. Although the latest automatic feed regulators are practically perfect, the person in charge of the feeding arrangements must be cautioned that careful attention is necessary in observing the water-gauge glasses for any signs of shortness of water. If the water disappears altogether the draught plate doors should be shut, to check combustion, and if the water still remains out of sight the stop valve should be shut and the safety valve eased. The fires can then be put out. In this connection it will be of benefit to describe the safety methods which should be adopted if a tube bursts when steaming. Shut and secure the ashpit doors, open the safety valve, increase the fan speed, increase the supply of feed water and start the auxiliary feed pump, making sure that the auxiliary feed check valve is open, shut the stop valves, and station a man to prevent the stokehold hatches from being opened.

Addition of Lime.

When under way, lime in solution should be introduced into the feed water in all types of water-tube boilers. Besides preserving the boilers, this lime should maintain the boiler water in a slightly alkaline condition. The amount of lime necessary to keep the water in an alkaline state should never exceed 2 lb. per 1,000 h.p. per 24 hours. In the case of the Belleville type 3 lb. should not be exceeded. The amounts stated should easily maintain the necessary alkalinity; if not, a leakage of sea water may be inferred. Feed water should never be allowed to become acid, slightly alkaline being the ideal condition. The lime should not be added in large quantities, but must be introduced into the lime tanks at frequent intervals, and the discharge feed water so arranged that the lime is regularly and continuously discharged till the next charge is inserted. It is of the utmost importance to keep the feed water filters clean and in good order, so that no clogging of the filtering medium can occur. Feed filters are usually duplicated, so that the cleaning can be carried out without interruption. As a general rule, the filters will require cleaning after 80 hours' continuous steaming, but this, of course, will depend upon the quality of the feed water, cleanliness of the interior of the boiler, &c. &c. Water-tube boilers should be scummed and blown down every 12 hours, and also after fires die out. If the boiler is to be opened for cleaning it may be blown right out when the pressure drops to 20 lb. per sq. in. This method makes the cleaning of the interior of boiler and tubes a much easier matter. In Babcock & Wilcox boilers, where the greasy deposits are greater under the main feed inlet, the corresponding side will require to be blown down more frequently than the other; also boilers of this type should be blown down twice, and scummed one each watch.

Care of Boilers after Use.

When a boiler is finished with it should be blown down and scummed and the fires allowed to burn down slowly, so that the circulation in the boiler is maintained as long as possible. After steam is down the boiler should be pumped full, and a slight pressure maintained (about 10 lb. per sq. in. is usual). Should the boiler not be required for service or cleaning, this pressure should be maintained, and to prevent moisture forming on the external surfaces small stoves should be kept alight in the ashpits.

In cases where it is impossible to keep the boilers full of fresh water one of the two following methods should be

adopted: (1) Empty and thoroughly dry the boilers by means of airing stoves, and replace all doors except the lower drum doors. Into these drums small bogies of burning coke or charcoal are introduced, and the boilers are then hermetically sealed. This method ensures no internal decay taking place, as all the oxygen should be consumed. (2) Empty and dry as above. Trays containing quicklime are then placed in the lower and upper drums, and a tray of burning coal introduced into the boiler. This increases the efficiency of the lime by consuming at once as much as possible of the oxygen of the air inside the boilers. The boilers should then be closed.

If neither of the above methods are practicable, the boilers may be kept empty and warm. This is done by keeping small bogies burning for a few hours daily in the ashpit. Care must be taken that all parts of the boilers are warmed equally.

Examination of Water-tube Boilers.

External.—Boilers should be cleaned and examined externally after fires are out and all ashes and dust removed, and also before a boiler is opened out for internal cleaning and examination. The most important parts are, of course, the tubes and tube ends, and the fire rows need most careful sweeping and scraping, and the ends of the fire rows carefully examined for signs of leakage. Any signs of leakage should be dealt with at once, the defective tubes being re-rolled or renewed. The tubes should be examined next for distortion. Slight distortions in curved tubes are easily straightened in place, but where this does not affect the steam tightness, nor obstruct the passage of gases through the tubes, it is immaterial and may be disregarded. If the distortion exceeds 1 in., both in straight and curved tubes, the tubes should be attended to. When curved tubes are straightened in place extreme care should be taken that the tube holes in tube plates are not distorted. Small changes in curvature in a Yarrow boiler are unimportant, but in the Niclausse and Dürr boilers tubes with a curvature exceeding $\frac{1}{2}$ in., the inner tube should be removed and bent to the curvature of the outer. In straight-tube boilers special gear is provided for straightening the tubes, and this operation should be carried out, if necessary, at each thorough clean. A good plan is to try the fire rows with a wooden batten each time the boiler is finished with after a period of steaming.

Drums.—Water and steam drums must be examined periodically on the outside for any signs of corrosion, roughness, &c. The best plan is to have the lagging for these parts made in portable pads, wired on, it then being a simple matter to remove the pads when required. All drums must be painted externally with red lead paint.

Brickwork.—It is very essential that the brickwork in water-tube boilers is at all times kept in an efficient condition, especially the parts in contact with boiler parts. The furnace door arch brickwork is very important, and must at all times be kept in perfect repair in order to guard against injury to casings, lower tube joints, and where soot, dirt, &c., is likely to accumulate.

Internal examination and cleaning.—The chief cause of uneconomical working in all water-tube boilers is a dirty condition of the internal surfaces of the tubes and steam and water drums. Boilers must be cleaned internally, and all tubes cleaned and searched at regular intervals. We will first consider a partial examination. This should be made when time does not permit of the more extended inspection, although it gives a good general idea as to the internal condition.

Remove doors of the two fire rows and examine the tubes (electric searcher and mirror). Should no undue deposit be seen, remove a diagonal row of tube doors right across the boiler from top to bottom or *vice versa*. This method will give an excellent idea as to the condition of each row. Remove one end door of steam collector, and two or more on the feed collector, and top and bottom doors of sediment and float chambers. If the deposits are soft they can be removed with the brushes supplied, but if signs of scale are found, the boiler should be boiled out with a strong soda solution. This boiling must be continued for at least 12 hours, and the proportion of soda should be 20 lb. per ton of water. During the boiling out the boiler should be blown down about every three hours, and, if scumming arrangements are fitted, scummed at the same time. After the boiling out, the fires are drawn, and the boiler allowed to cool. When steam is out, and the water sufficiently cool, run down below the steam drum and clean same. Keep lowering the water in the tubes, cleaning each row as the level falls. After all the tubes are cleaned in this way, run boiler right out, and complete cleaning the remaining part. In all cases when the zinc slabs are not removed they must be carefully examined in order to see that they are properly fastened to the hangers, and that good contact is being made. After a boiler has been cleaned and closed up it should be tested to the ordinary working pressure. The above procedure applies, of course, to straight-tube boilers. In boilers of the Yarrow type, when time does not admit of wire scrubbing the tubes internally, remove steam and water drum doors, search each tube with a flexible wire searcher, wash out with fresh water, and close up, testing as previously described.

Thorough Examination and Cleaning.

We will consider the large tube type first. Remove the following doors :—Steam collector end door, top and bottom doors of elements, at least two feed collector doors, top and bottom doors of float sediment chambers, at least an additional door from each element. Doors are removed diagonally, and the doors so removed must be noted, so that at subsequent cleanings and examinations different doors are taken off. Thus all the doors in the boiler will eventually have been removed and the tubes examined. Steam collector drain plugs must be removed. The internal fittings in the steam drum are removed and refitted, as are also the zinc slabs and their attachments. It will be found in practically every case that the slabs require renewal. The following fittings must be carefully examined :—Down-take pipe, non-return valves, connecting pipes from elements to tops and bottoms of water-gauge fittings, automatic float gear, automatic floats (ascertain that these are not water-logged), element supports, all casings, sediment chambers, blow-down fittings. At a thorough examination the lower rows of tubes must be gauged for straightness. Electrically-driven wire brushes are of great assistance in cleaning tubes internally.

Boilers of the Yarrow and small-tube type.—Remove one steam collector door and the water drum doors. Examine such internal fittings and casings previously described as apply to this type of boiler. First wire-scrub the steam reservoir, special care being taken at water line and round the various orifices. The tubes are next searched and cleaned with wire brushes on flexible handles, finishing off with soft brushes. As each tube is searched and cleaned it should be chalked at the end to avoid going over the ground twice. Finally, the water drums are wire-scrubbed and swept out. The boiler should be tested in the same manner as the large-tube type. Should any internal surface show signs of corrosion, blacklead should be applied and well brushed into such parts. Some engineers blacklead and polish the whole internal surface of steam and water drums, and this method gives excellent results. Another method, where corrosion appears in the steam drum above the heating surface, is to coat the parts with Portland cement mixed with a soda solution, but this must never be applied below the water level. With water-tube boilers, or, in fact, any high-pressure generator, paint, oil, grease or inflammable liquids must never be used on the interior surfaces. When corrosion appears, a good plan is to make a rough sketch showing details of such corrosion, so that any progress may be noted at subsequent examinations.

MARINE DIESEL ENGINES—RUNNING CONDITIONS AND OPERATING SYMPTOMS*—III.

Starting and Manœuvring—Starting Momentum.

(*Continued from page* 71.)

Starting and Manœuvring.—The great problem under this heading is conservation of blast and starting air.

A six cylinder four stroke engine, has, regardless of position, always got at least

(*a*) One cylinder on the firing or expansion stroke.

(*b*) One cylinder on the compression or firing stroke.

(*c*) One cylinder on the suction or compression stroke.

Now, even under the worst condition, all the air that is needed to start the engine is for one impulse to be given to cylinder No. 1, followed by an impulse to cylinder No. 5, since the piston in cylinder No. 3 is rising on the compression stroke and will receive a charge of fuel about top centre which will effectively keep the engine in motion until the next fuel valve opens in sequence. Should the fuel charge given to cylinder No. 3 fail to ignite, then, assuming that all air-starting valves have been cut out, another air impulse must be given, this time to cylinder No. 6, which will bring the piston in cylinder No. 12 to top centre, where it will receive a fuel charge. There is nothing to be gained by allowing the engine to make several revolutions with all starting valves in action. On the contrary, the effect is to cool all the cylinders and to introduce moisture, which passes with the starting air, and which acts as a great deterrent to the ignition of the fuel due to the heat absorbed in its evaporation.

Provided the cylinder head valves and piston-rings are in a reasonable state, the starting acceleration due to one air impulse is sufficient to raise the temperature on a single compression stroke, to a degree high enough to ensure ignition of the fuel charge. It has been the author's experience that an air pressure merely high enough to turn the engine without sticking is sufficient to ensure a start on fuel, and when it is considered that the pressure referred to is but 180 lbs. per sq. in., it will be gathered that the acceleration rate is not very high.

When an engine does not readily pick up on fuel, the most common reason is due to air in the fuel piping between the injection pumps and injection valves. The reatest care should, therefore, be taken to ensure that the process of "flooding through" is carried out thoroughly, as on this, more than anything else, instantaneous fuel starting depends.

It will be readily understood that the most economic consumption of starting air depends upon the period elapsing between the time the starting valve is cut out and the fuel valve is brought into operation, so that the time element plays an important part.

Considering now the blast air question. The principal saving in blast air is effected by manœuvring the engines with the lowest possible pressure, in cases where the fuel injection valve lift remains a constant, and by reducing the fuel valve lift and also the blast pressure in cases where variable injection valve timing is provided. For the best manœuvring conditions, as well as the greatest flexibility, a variable gear is essential.

When manœuvring an engine it is considered desirable to empty the blast air pipes between the control valve and the fuel injection valves as soon as the engine is stopped, since there is a danger of any air that remains in the fuel injection valves or blast pipings finding its way past the fuel non-return valves into the fuel piping and so causing an air lock, which might even extend to the fuel injection pumps. It is this source of loss that causes the greatest consumption of blast air, since the pipe line and fuel valves have to be refilled each time a start is made. The author is rather of the opinion that this emptying of the system should be abolished, and that more trust should be placed in the efficacy of the fuel non-return valves. Experience with auxiliary engines, where draining of the blast piping is more the exception than the rule, has gone to show that an engine can be started after lying for a whole day with blast air in the system, without needing to be "flooded through" again.

Apart from the loss incurred through draining, the consumption of blast air is not at all high when manœuvring, especially where variable valve gear is fitted. By "linking-in" to use a steam-engine expression, or, in other words, by reducing the lift of the fuel valves, a considerable saving of air is effected.

With all engines, the aim is to make as much blast air as possible, to use the least quantity for injection purposes, and to pass the surplus to augment the starting air supply. Where a variable gear is not fitted, this is often done at the expense of good combustion and with dire results to the fuel injection valves in particular. The blast pressure is reduced to the lowest possible figure, about 560 lbs. per sq. in., and while combustion remains normally at constant pressure all is well, but should it change to approximately constant volume with an accompanying pressure rise, then there is the danger of an explosion taking place in the fuel valve. Another effect is, that part of the products of combustion are forced back into the amotizer space and tend to choke the slots or holes. This choking is not noticed at the time, unfortunately, but as soon as "full away" is received it is found that the defective valve will not pass the necessary quantity of oil in the given time, thus the speed falls due to late injection. In cases where the choking is great, the engine has to be stopped and the valve changed.

Where variable valve gear is fitted, an important point to observe is that the fuel valve lift is at all times strictly proportioned to the quantity of fuel being delivered by the pumps. In executing telegraph orders for slow speed, the necessary reduction in revolutions is often got by merely "linking-in" without first reducing the amount of fuel being supplied by the pumps. The effect is, that the oil, being supplied at a greater rate than the reduced lift can cope with, builds up in the air space of the fuel valve. Should the slow speed order prove to be of long duration the fuel will find its way into the blast air piping where its presence is dangerous. Again, should a demand for increased speed be received and the valve lift increased accordingly, a violent explosion in the cylinder will result due to the excess oil passing in bulk. Even in the worst case of operating at slow speed, detonation will occasionally occur due to this same cause. The most probable reason is that the fuel tends to enter the cylinder in a solid state instead of being atomised—the depth of oil lying in the valve becoming too great for the blast air to properly penetrate it, so that atomisation is impossible.

The most advisable procedure, when operating an engine with very reduced fuel valve lift, is occasionally to back the gear into mid position and so allow the fuel valve to clear itself. In doing this, the revolution speed will rise, but if the gear is immediately returned to its original position the momentary increase will prove negligible.

(*To be continued.*)

* Abstract of paper read by Mr. A. J. Brewer, before the Institution of Marine Engineers.

NOTES FROM THE FOREIGN PRESS.

Specially abstracted for "The Marine Engineer and Naval Architect."

Electric Resistance Welding in Shipbuilding.

Electric welding has been developing in many directions and applications of late. In America and England the arc weld is favoured, but in Germany the resistance weld is being greatly adopted. Spot welding is used in many of the sheet plate industries and with electrodes under a pressure of 8,000 kg./cm. plates up to 12·5 mm. have been welded. However, in such work where watertightness as well as strength is required, attention was early directed, and especially in America, to the possibilities of lap welding. The American progress is said to have been hindered by the tendency of the electrodes to fire when the surfaces to be welded have not been free from dirt. In the German machines this has been overcome by making the roller electrode matter discontinuous. As far as the motion of the roller is visible to the eye it is apparently moving continuously. With this machine welding of steel, copper, zinc, aluminium, &c., has been carried out without any previous treatment of the metal surfaces. Electric-butt welding is also being successfully applied. One interesting application being the manufacture of steel chains. The material forming the chains is heated, bent to shape, and welded electrically. The electric hearth or furnace is being increasingly adopted and is replacing the ordinary coal furnace. The electric rivet-heater is now quite well known in shipbuilding. The electric-resistance welding and resistance processes offer considerable scope for the reducing of the costs of many industrial processes. It is claimed that they are at least 20 to 80 per cent. more economical than those involving ordinary coal methods of heating. The development of electric heating and welding in shipbuilding should thus be rapid, and, as many yards produce their own power, their existing plant can be thus much better utilised. Economy and attention to fuel consumption can be carried out in a power station, but with present methods of coal heating scattered about a ship yard, economy is difficult to effect.—*D. Neuburger, Schiffbau*, December 20, 27, 1922, p. 198 ; 3 pp. ; 5 ff.

Contrary-Turning Co-Axial Screws.

In this article General Rota reviews the present position of the subject of contrary-turning co-axial screws and especially from the point of view of their adoption in merchant ships. The early development is outlined and reference is made to the work of Ericsson, Rigg, Thornycroft, Berkett and Wagner. The theoretical work of Greenhill on the subject is touched upon to show that increased efficiency can be expected with co-axial screws ; and the recent adoption of this type of propulsion in aeronautics is mentioned as a further illustration of their practicability of promise. The experiments carried out in 1909 by General Rota on a steam launch are mentioned, and from them it is shown that a saving of power of 20 per cent. over a single-screw installation could be effected by using contrary-turning co-axial screws. Next, Luke's experiments are outlined. With co-axial screws it was found that whilst an average increase in hull efficiency of 20 per cent. could be obtained over single-screw and twin-screw installations, yet the efficiency of the propellers themselves was only 86 per cent. of that of a single screw. From this result Mr. Luke concluded that, in view of the mechanical difficulties attached to co-axial screws, their slight increase in overall efficiency was not sufficient to justify their adoption. Finally, General Rota deals with his own most recent experiments. In these, Luke's experiments were confirmed, but it was also shown that it was unfair to generalise from them. When other screws were used of different diameter and pitch, it was found that the efficiency of the co-axial screws and the single screws in the open was practically identical. Thus, in the overall efficiency, the full saving in hull efficiency is in the favour of the co-axial screws. Finally, General Rota concludes by recommending the adoption of this type of propulsion in merchant ships. With engines aft, the mechanical difficulties ought to be easily overcome, and the possibility of an average increase of overall efficiency of 20 per cent. is not to be lightly rejected.—*G. Rota, La Marina Italiana*, January, 1923, p. 15 ; 5 pp. ; 2 ff.

Air Screws for Air and Water Craft.

This article forms the first of a series to be devoted to the publication of the results of research work carried out by Dr. Schaffran on air screws at the Berlin ship experiment station. The experiments were carried out in water and form a very comprehensive series. All the model screws were made of bronze and have a diameter of 120 mm. The whole series is classified as series B. The first group of the series embraces two-bladed screws of constant pitch and pitch-ratio varying from 0·4 to 1·2, having header-width-ratios of 7·5, 10 and 12·5 per cent. respectively, and a constant thickness-ratio of 2·5 per cent. All ratios, of course, refer to the diameter. Complete diagrams are given of all the screws in the series. The second group of the series is identical with the first, only the propellers are four-bladed. The third group is formed from the 7·5 per cent. blade-width-ratio screws of the first, the screws being used as contrary-turning co-axial screws. The fourth series is similar to the third, only the direction of rotation is identical instead of contrary, *i.e.*, tandem screws. The fifth and final group is similar to the first and second, only the screws are three-bladed. In the actual experiments the screw axis was, for the whole of the series, immersed one diameter, *i.e.*, 120 mm. A constant number of revolutions of 30 per sec. was adopted and the translational velocity varied from zero to a maximum, thus covering the range of slip and thrust likely to be met with in practice. The propeller dynamometer is of an improved type, one special feature being that the length of shaft to which the propeller is attached is much greater than usual. By use of concentric shafting the dynamometer can be converted to carry out the co-axial contrary-turning series. With this series, whilst the two propellers can be tried over a range of separation, each screw must necessarily have the same number of revolutions. For the tandem screws only a single shaft is necessary and these again, whilst having a range of separation, can only be run at the same number of revolutions. (*To be continued.*)—*Dr. K. Schaffran, Schiffbau*, February 14, 1923, p. 319 ; 6 pp. ; 7 ff.

Service Results of the T.S.M.S. "Zoppot."

The twin-screw motorship *Zoppot*, built at Krupps, Germania-Werft, Kiel-Gaarden, in August, 1920, for the Hamburg-American Petroleum Company, is a tanker of 15,750 tons d.w., 10½ knots speed, fitted with twin two-stroke Diesel engines, also built by Messrs. Krupps. For two years the vessel has been chartered by the Standard Oil Company, and in December last returned to Hamburg and thence to Kiel to have several minor alterations carried out. The engines have each a b.h.p. of 1,750 an i.h.p. of 2,360 and run at 105 r.p.m. The author of this article, Ober-Marinebaurat Laudahn, has had the whole of the vessel's logs placed at his disposal and he is thus able to give a very detailed description of the behaviour of the vessel's machinery on service. Each part of the machinery is commented on in turn, and the lessons of the vessel's experience are well brought out. The vessel's performance shows that the two-stroke is as reliable as the four-stroke, and is such that the adoption of a twin-screw installation from the point of view of safety is quite unwarranted. Much detailed information and many illustrations are given in the article. The average fuel consumption (fuel not specified) of the main machinery was 0·304 lb. per i.h.p.-hour. The lubricating oil consumption was 0·00255 lb. per i.h.p.-hour. The fuel and lubricating oil costs, per 1,000 ton-miles, work out at 5·63 c. The price of oil is not specified. The *Zoppot* proves undoubtedly the reliability of the two-stroke Diesel-engine for marine propulsion.—*W. Laudahn, Schiffbau*, January 10–17, 24, 31, 1923, pp. 231, 255, 276 ; 17 pp. 21 ff.

Producer Gas Installations and French Colonial River Vessels.

Development of producer gas installations in land work directs attention to their application for colonial river navigation, where hitherto steam and vegetable oil motors only have been used. The excessive weight of the steam plant, together with the excessive amount of pull required, greatly diminshes the deadweight capacity of the steam-driven vessel. Vegetable oil motors, whilst much more

oil engine, suffer from the fact that a high price can be obtained for the oil at the ordinary market and thus competition will again increase the price. In this article the possibilities of producer gas are considered. The technique and lay-out of the plant is outlined. The producer gas plant is shown to be much more efficient than the ordinary steam plant and has a fuel consumption of 500 grams of wood charcoal per h.p.-hour. Its only serious rival is, of course, the vegetable oil motor, whose consumption is 250 grammes per h.p.-hour. This latter is lighter than the gas installation and is more convenient in service. However, the producer gas plant is in use on numerous German craft, and although the number in France is much smaller, at least one motor barge, the *Maria Theresa* has been in successful service since 1914. The article gives a useful review of the possibilities of the subject.—*Bulletin Techn. du B.V.*, January 1922, p. 13, 3 pp.; 3 ff.

Ship Breaking.

Before the war the shipbreaking industry was comparatively small and unremunerative. Since the war, especially in Germany, the industry has been transferred into one of considerable importance. The lay-out of a shipbreaking yard differs in many respects from a shipbuilding yard. Much greater quay space must be provided and numerous quay cranes of considerable lifting power must be available. The quantity of material transported is much greater than in a shipyard and the railway facilities must be carefully organised. In dismantling a ship the oxy-acetylene cutter is absolutely indispensable. The largest possible plates are cut out by the flame and are afterwards cut down by shears to a size suitable for the steelworks. Oxy-acetylene generation is one of the main expenses of a shipbreaking yard, and to reduce the expense of transportation many firms have their own gas producing plants. The supervision of employees is an important aspect of this industry. The presence of such dangerous gases and carelessness of the workmen require elaborate regulations to be framed and respected. As much of the dismantled contents of a ship are readily disposable and transportable, considerable precautions have to be taken to guard against pilfering. As most material is sold by weight a weigh-bridge is attached to most yards and is found to be absolutely essential as a check against the weights stated by the railway company. In the dismantling of ships quite a number of precautions must be taken. For instance, the structure should be preferably dismantled horizontally both from a point of view of securing adequate lighting and ventilation as well as to preserve sufficient strength of structure to resist possible deforming forces. Care has to be taken that the material is not taken away too near the waterline in case the vessel should heel and sink through the admission of water. Many other interesting features of this industry are dealt with in this article.—*Momber, Werft. Reederei, Hafen*, December 7, 1922, p. 766; 4 pp.; 7 ff.

Stability and Safety of Passenger Vessels.

Much technical consideration has been bestowed upon the important subject of ship safety since, and as a result of, the loss of the *Titanic*. In this article the development of the subject since that date is reviewed. The report of British Bulkhead Committee is critically examined with regard to stability recommendations. A statement made by Sir Archibald Denny in discussing Brimblecombe's 1914 International Navigation Association paper, viz., "That there was no known case of a vessel having been bilged in service and then capsizing," is commented upon in the light of subsequent experience with the *Empress of Ireland*, *Lusitania* and *Egypt*. In all these cases the vessels heeled so much that the lowering of the boats was either entirely prevented or seriously interfered with. The pre-war views of Dr. Foerster and Prof. Flamen are also commented on in this light, and Foerster's development of the "Formstabil" type of vessel such as the *Cap Polonio* and *Albert Ballin* is mentioned to show that the importance of stability is being increasingly recognised. After further considering the stability problem, the author outlines a proposal for the better and safer construction of passenger vessels. The principal idea of the system is to flare out the vessel's topsides amidships. By this means it is claimed that boats may be much more widely lowered and the lowering gear simplified. The projecting decks will minimise the danger of collision and will tend to confine damage to the top structure, whilst the longitudinal structural weight is increased by about 0·7 per cent. in a vessel which retains the usual high erections. However, the extra beam and deck structure brings about an increase in the longitudinal strength of the vessel and thus the structural weight could be reduced. The wide deck beam is very suitable for accommodation purposes and will tend to preserve dry decks in rough weather. Whilst the vessel may cost slightly more than the usual type, if full advantage is taken of the various possible improvements in the design stage this increase ought to be insignificant. The proposed vessel is fully described and illustrated, and a detail comparison and calculation of the structural weight is included with the article.—*L. Chiradi, La Marina Italiana*, January, 1923, p. 6; 9 pp.; 4 ff.

Tubular Ships.

Starting from the well-known geometrical fact that for a given perimeter the circle has the greatest area of any figure, Signor Vitulli-Montanuli, a shipowner of Genoa, has produced a very economical type of ship. For the same perimeter of cross-section, and hence weight of shell, a circular form can carry 25 per cent. more displacement than a box form. The circular form is ideal from the point of view of wetted surface, and thus as frictional resistance forms the greater part of a vessel's resistance, the engine power for the same speed can be reduced. Viewed from another standpoint, it can be shown that a tubular ship can carry for the same hull weight, 26 per cent. more deadweight than a box form. For the same deadweight the tubular ship has 20 per cent. less hull weight than a box form. For the same displacement a tubular vessel has about 16·6 per cent. less hull weight and 4·7 per cent. more deadweight than a box form. It is further claimed that the tubular type gives a more rational distribution of material from the structural strength standpoint. From these considerations the twin-screw motorship *Adriana* was constructed. The following are its main particulars:—Length between perpendiculars, 103·5 m.; beam, 15·5 m.; depth, 10·83 m.; deadweight, 6,104 metric tons; displacement, 9,104 metric tons; draught, 7·410 m.; wetted surface, 2,450 sq. m.; block coefficient, 0·749. The machinery is formed of two 1,000-b.h.p. sets of Ansaldo-San Giorgio two-stroke heavy oil engines, which at 10·5 knots consume 9·5 tons of fuel per day. Complete diagrams accompany the article. The weight of the *Adriana* is unfortunately not given, but it is mentioned that full advantage of weight reduction has not been taken because of the severe trade that this special vessel was intended for. The construction is approved by Lloyd's Register, British Corporation, Registro Italiana and Bureau Veritas.—*M. Codebo, La Marina Italiana*, January, 1923, p. 20; 5 pp.; 4 ff.

Lloyd's New Rules.

This article forms an interesting criticism and review of Lloyd's new rules. The following points are dealt with in brief detail:—Side plating, double bottoms, elimination of side stringers, bow strengthening, and increase of frame spacing. Whilst the transference of material from the side plating to the decks is one of the commendable features of the new rules, the reduction of plate thickness in way of the waterline where corrosion is greatest is not quite clear to the author. He presumes, however, that this step is backed up by the experience of Lloyd's. In the construction of the double bottom the author observes that Lloyd's are following British Corporation in the matter of simplification. In the eliminating of parting beams and the stiffening of frames forward, the author points out that Lloyd's are following the pre-war practice of the Germanischer Lloyds. With regard to frame spacing, the author considers that Lloyd's are giving a useful lead to other classification societies both in the way of increased frame spacing and in the elasticity of rivet spacing to admit of simple and standard production. Finally, the author criticises to some extent the frame numeral of Lloyd's. He points out that draft does not yet enter into the determination of frame scantling in Lloyd's rules, and he considers that Lloyd's should further follow British Corporation in this respect. In discussing the second longitudinal numeral (B D) he points out that as a rectangular figure of given area has a minimum perimeter when it is square, that Lloyd's numeral encourages the building of deep ships. The article is one of a type which is all too rare, and forms an interesting commentary on classification society regulations.—*V. Baucer, La Marina Italiana*, February, 1923, p. 45; 2 pp.

G.E.C. Research Laboratories.—We have had, just before going to press, an opportunity of inspecting the research laboratories of the General Electric Company, at Wembley, which were opened on February 27. In view of our Leading Article dealing with the Combination of the Talents, the opening of such a department strikes us as decidedly one of the steps in the direction proposed in the article. The laboratories are well fitted, and their general lay-out should secure efficiency of working. The object of the laboratories is to aid the scientific production of the various electrical appliances manufactured by the General Electric Company, so that processes in manufacture, which before were carried out in certain stages more or less by guess work and by rule of thumb, can now be carried out on a definite scientific basis, with the immediate rectification of errors. Among others, the question of electric lamps is dealt with very thoroughly.

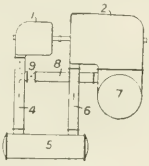
RECENT PATENTS RELATING TO MARINE ENGINEERING.

Compiled by E. P. Alexander & Sons, Chartered Patent Agents, 306, High Holborn, London, W.C.

No. 188,877. Turbines.

Baumann, K., Northwood House, Barnfield, Urmston, Lancashire, and Metropolitan-Vickers Electrical Company Limited, 4, Central Buildings, Westminster. October 4, 1921, No. 26231. [*Class* 110 (iii).]

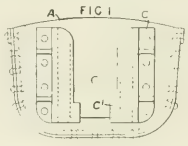
Relates to a by-pass device for preventing acceleration or racing of steam turbines after the governor valve has been closed, arising from steam leaking past the valve or from steam remaining in the system after the valve closure. In an installation comprising high and low pressure turbines 1, 2 and having a reheater 5 in the intermediate piping 4, 6, a by-pass 8 controllable by a valve 9 operatively connected with the main or emergency governor is adapted to open simultaneously with or promptly after the closure of a governor valve and lead the leakage or residual steam direct to the condenser 7. The by-pass may connect any suitable part of relatively high pressure with one of low pressure.



No. 188,966. Furnaces.

Clement, W. E., Hillrise, Parc Howard Avenue, and Thomas & Clement Limited, Morfa Engineering Works, New Dock, both in Llanelly, Carmarthenshire. November 28, 1921, No. 31844. [*Class* 51 (ii).]

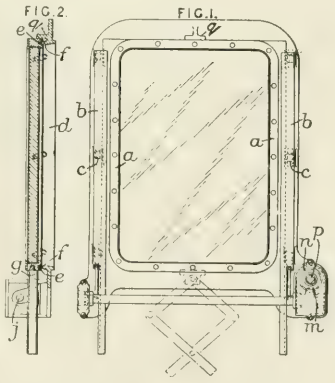
Charging.—A charging-box for steel furnaces is formed with a pressed steel end plate A riveted to the trough-like body of the box, and to this end plate are secured Z or similar strips C which form the socket for the head of the charging-arm. The stop for the head may be formed by inturned portions C^1 of the strips C, or by a plate secured to or formed integrally with the strips. The plate A and the strips C may be curved to suit the radius of swing of the charging-arm.



No. 188,993. Ships' Windows.

Swan, Hunter & W. Richardson Limited, Wallsend-on-Tyne, and Temple, G. T., 35, Grosvenor Drive, Whitley Bay, Northumberland. December 16, 1921, No. 33952. [*Class* 20 (iii).]

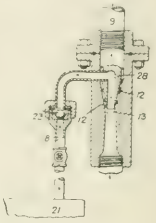
In vertically sliding windows, and particularly ships' cabin and deck windows, the glass carrier *a* slides on a frame *b* connected by parallel pivoted links *c* to a fixed frame *d* and movable relative thereto by worm and pinion gearing *n*, *m*, so as to engage the jointing ring *e* and rib *f*. The frame *b* is operated by links *k* keyed to the pinion shaft *j* and connecting links *g* pivoted to the frame. The worm is operated by a key engaging the squared end of the spindle *p*. The window is of the balanced sliding kind, and a stop *q* is provided for the highest position.



No. 189,770. Ejectors.

Kobiolke, A. M., Gilderoy, Yarra Junction, and Giffney, A. J., 317, Collins Street, Melbourne, both in Victoria, Australia. November 13, 1922, No. 31005. Convention date, December 5, 1921. Not yet accepted. Abridged as open to inspection under Sect. 91 of the Act. [*Class* 71.]

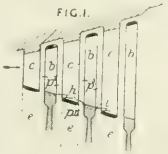
In apparatus for removing air and other gases from vessels the inducing fluid is caused to form a hollow whirling jet or vortex into the hollow of which the air, &c., is drawn. Inducting fluid entering by a pipe 9 is forced to take a spiral course through the annular converging nozzle formed between the inner and outer converging cones 28, 12 by reason of a continuous baffle or rib extending completely across the annulus. The inner cone ends shortly before the throat 13 of the receiver, and the induced fluid passes from the vessel 21 to be exhausted through a non-return valve 23, and a lateral pipe 18 to a continuation thereof centrally within the inner cone of the nozzle.



No. 189,457. Turbines.

Akt.-Ges. Brown, Boveri, et Cie, Baden, Switzerland. November 21, 1922, No. 31845. Convention date, November 26, 1921. Not yet accepted. Abridged as open to inspection under Sect. 91 of the Act. [*Class* 110 (iii).]

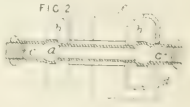
In order to dispense with, or reduce the number of, the fixed partitions separating the pressure stages of a multi-stage impulse turbine, the inlet and outlet edges of the stator guide blades *c* terminate nearer the shaft than the edges of the adjacent wheel blades *b*, with the result that the motive fluid entering the guide-blades *c* has a suction action on the space *e*, and on passing out has a pressure action on the space between the two wheels, thereby equalising the pressure throughout the space to a value p^{11} lying between the values p^2 p^3 on either side of the guide-blades, and obviating the need for a partition. The wheel rim may be provided with projections *h*, *i* for deflecting the stream of motive fluid towards and away from the guide members. The axial thrust is taken by a thrust bearing, or by a balance piston, or by a duplex arragement of the turbine. Impulse stages without partitions may be combined in the same casing with either impulse stages fitted with partitions, or a reaction turbine of the kind described in Specification 175,971, or with both. As described in Specification 124,734 in connection with reaction turbines, the guide blades may be made much wider than the wheel blades to permit of increased width for the wheel hubs.



No. 189,713. Boilers.

Pearce, F. R. R. J., 96, Nimrod Road, Streatham, London. March 31, 1922, No. 9288. [*Class* 64 (i).]

A circulating boiler ring *a* with inlet and outlet, *c*, c^1 is supported by uprights *b* over a gas ring, and is provided with a heat-retaining shield *d* on which utensils may rest over the opening *e*. A number of such ring boilers may be connected together according to the number of gas rings available.

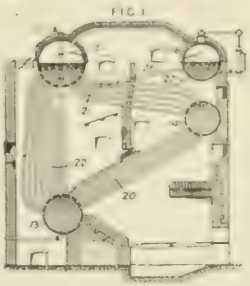


No. 189,447. Steam-generators.

Kidwell, E., 288, East Water Street, Milwaukee, Wisconsin, U.S.A. November 15, 1922, No. 31241. Convention date, November 22, 1921.

[illegible] 170,118. *Not yet accepted.* [*Accepted as open to inspection under Sect. 91 (3) (a).*] [*Class* 123 (ii).]

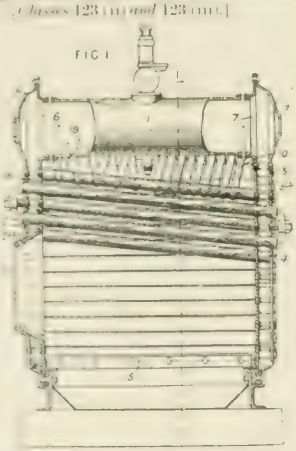
In a boiler having transverse drums 13, 16 so connected as to maintain a water circulation through the tube banks 20, 21, 22, and a supplemental circulation through the drum 16 from which the steam is taken, as described in the parent specification, the water in the supplemental circulation flows through two oppositely inclined tubes or groups of tubes 26, 37, connecting the drum 16 with the steam drum 15. The tubes may have upwardly curved ends, or they may be straight throughout their lengths.



No. 189,699. Steam-generators.

Wiley, S.E., 17, *Ebbleheigh House, Caxton Street, Westminster. February* 10, 1922. *No.* 3953. [*Classes* 123 (ii) *and* 123 (iii).]

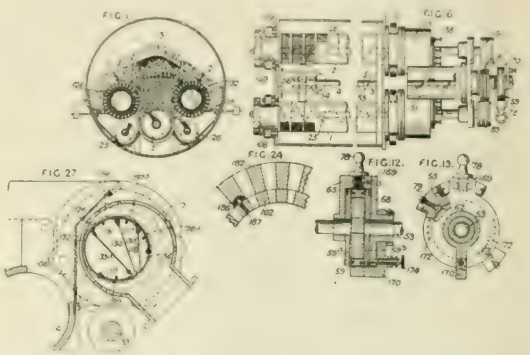
A water-tube steam generator comprises in combination front and rear headers each comprising a pair of plates 2, 2a, 3, 3a, pressed to shape to form a single water space, an upper horizontal steam and water drum 1 mounted with its ends directly in the inner plates of the headers at 6, 7 and its end walls constituted by the upper parts 8, 9 of the outer plates 2, 3 of the headers, inclined water tubes 4 extending across and above the fire-grates 5, a firing opening 11 formed through one of the headers, and an outlet or outlets for the products of combustion formed through the other headers. The outer vertical rows of tubes are pitched closely to form the walls of the combustion chamber, and they are enclosed in protecting tubes. The tubes may be expanded into the inner plates of the headers, or pass through those plates to the outer plates in which case they have openings 16 for communicating with the headers. Two superheater coils 19 constitute the flues leading to the opening in the rear header. Hot gases pass radially through the widely spaced coils at the forward end and then pass axially inside the closely spaced coils at the rear end.



No. 188,822. Furnaces.

Beaucaples, C. W., 211, *McGill Buildings, Montreal, Canada. August* 29, 1921. *No.* 22915. [*Classes* 51 (ii) *and* 80 (ii).]

An underfeed furnace comprises rotary grates operating in directions towards the side walls of the fire-box, and fed with fuel from a central screw feeder. Air for combustion is led to stationary chambers within each grate and is discharged to different areas of the fire-box. The furnace is shown applied to a Lancashire boiler, and comprises two grates 1, 2 arranged at the sides of the fuel-feed trough 4, so that the rotation of the grates assists in the feeding of the fuel and carries the ashes to the sides of the fire-box, depositing them on the counter-weighted plates 17, 18. The ash-pits are formed below the grates and are furnished with screw conveyors 25, 26, driven, together with the fuel-feed screw 5, from the front of the furnace. The rear ends of the ash-conveyor shafts carry pinions 108 engaging gear wheels 109 on the cylindrical grates, thus driving the grates at a speed in accord with the rate of fuel-feed. The trough 4 extends from the hopper 51 to an upwardly-sloping back-plate, behind which is the rear bearing 54 for the conveyor shaft 53, which carries a screw of decreasing diameter from front to rear. The grates comprise cylindrical fire-bar sections, the bars 182 extending from a solid ring and having bevelled ends fitting into the undercut ends of an adjacent section. In a modification the solid ring may be formed centrally of a bar-section, and retained by a key 186, secured to the slotted inner cylinder 102 by a screw 187, as shown in Fig. 24, the slots in the inner cylinder registering with the openings between the bars. Air for combustion is delivered to a casing 113 at the front of the furnace and passes thence through openings 196, 197*a*, 198*a*, 199, controlled by dampers 116, 119 on the casing 132, which fits closely inside the shield 126 extending along the length of the grate. Each damper is controlled by a handle 138 and admits air to one of the chambers 133*a*, 134, 135, 136, thus enabling the amount of air delivered to each part of the fire to be

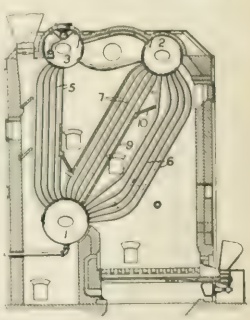


independently controlled. The grates are supported at the front and rear by rollers 111, 106, mounted in housings. The ash conveyors are operated by ratchet-wheels 89, 90 driven by eccentrics 81 on the fuel feed shaft 53. Adjustment of stroke and means for throwing the ratchets out of action are provided. The fuel-feed shaft is driven by the mechanism shown in Figs. 12 and 13. A ratchet-wheel 58 is fixed to the shaft 53 inside the housing composed of the side-plates 59*a*, 59*b* loose on the shaft, and the ring 59, carrying the lugs 179, inside which are spring-pressed pawls 62 engaging the wheel 58. A yoke made up of the arm 65 and plate 68 embraces the ring 59 and is provided with a boss 169 engaging one of the lugs 179. An arm 170, loose on the shaft, carries a plunger 174 engaging holes 172 in the plate 68 and allows variation of stroke. The yoke is provided with a ball 78 fitting into a universal coupling on a reciprocating driving rod.

No. 189,677. Steam-generators.

Ladd, G. T., 5,428, *Baum Boulevard, Pittsburg, U.S.A. January* 3, 1922. *No.* 167. [*Class* 123 (ii).]

In a water-tube boiler having a lower drum 1 connected to a front upper drum 2 by two banks of tubes 6, 7, and to a rear upper drum 3 by a single bank of tubes 5, a baffle 9 between the banks 6, 7 is so spaced away from the front bank that the furnace gases flow around this bank before entering the rear bank. A baffle 10 at the top of the baffle 9 deflects the gases against the upper ends of the tubes of the front bank. The gases pass downwards between the baffle 9 and a baffle 7 at the back of the rear bank and then upwards through the bank connected to the rear upper drum. The front upper drum is arranged somewhat below the level of the rear upper drum.



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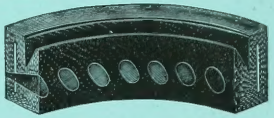

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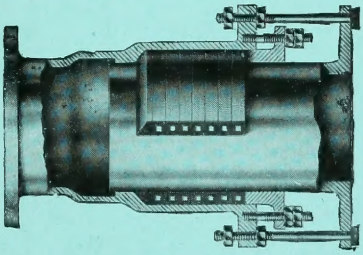

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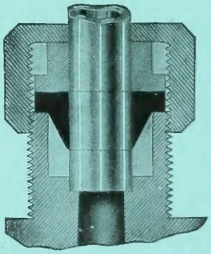

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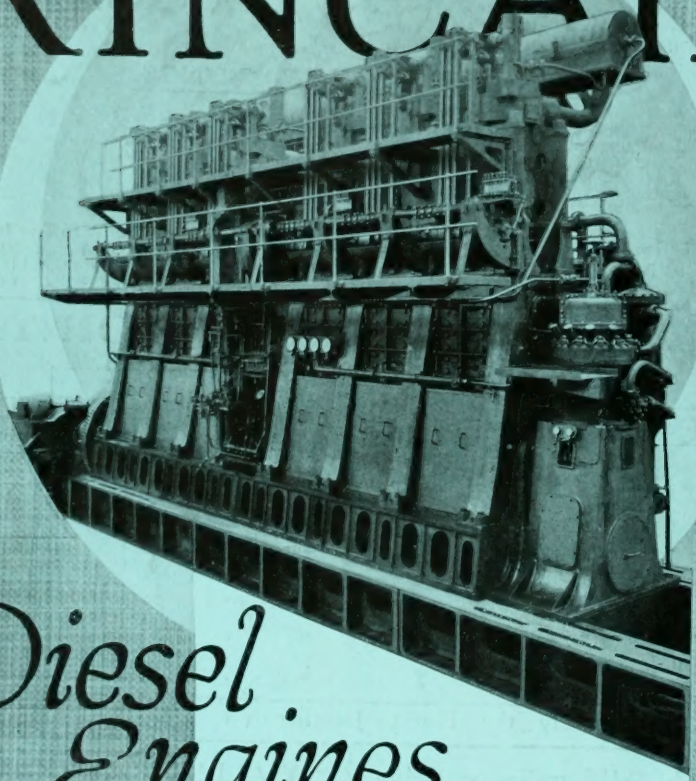

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London: Printed by Harrison & Sons, Ltd., Printers in Ordinary to His Majesty, 44, 45, 46 and 47, St. Martin's Lane, London, W.C. 2, and Published by The New Marine Engineer, Limited, Proprietors of The Marine Engineer, at 32, Tothill Street, Westminster, London, S.W.